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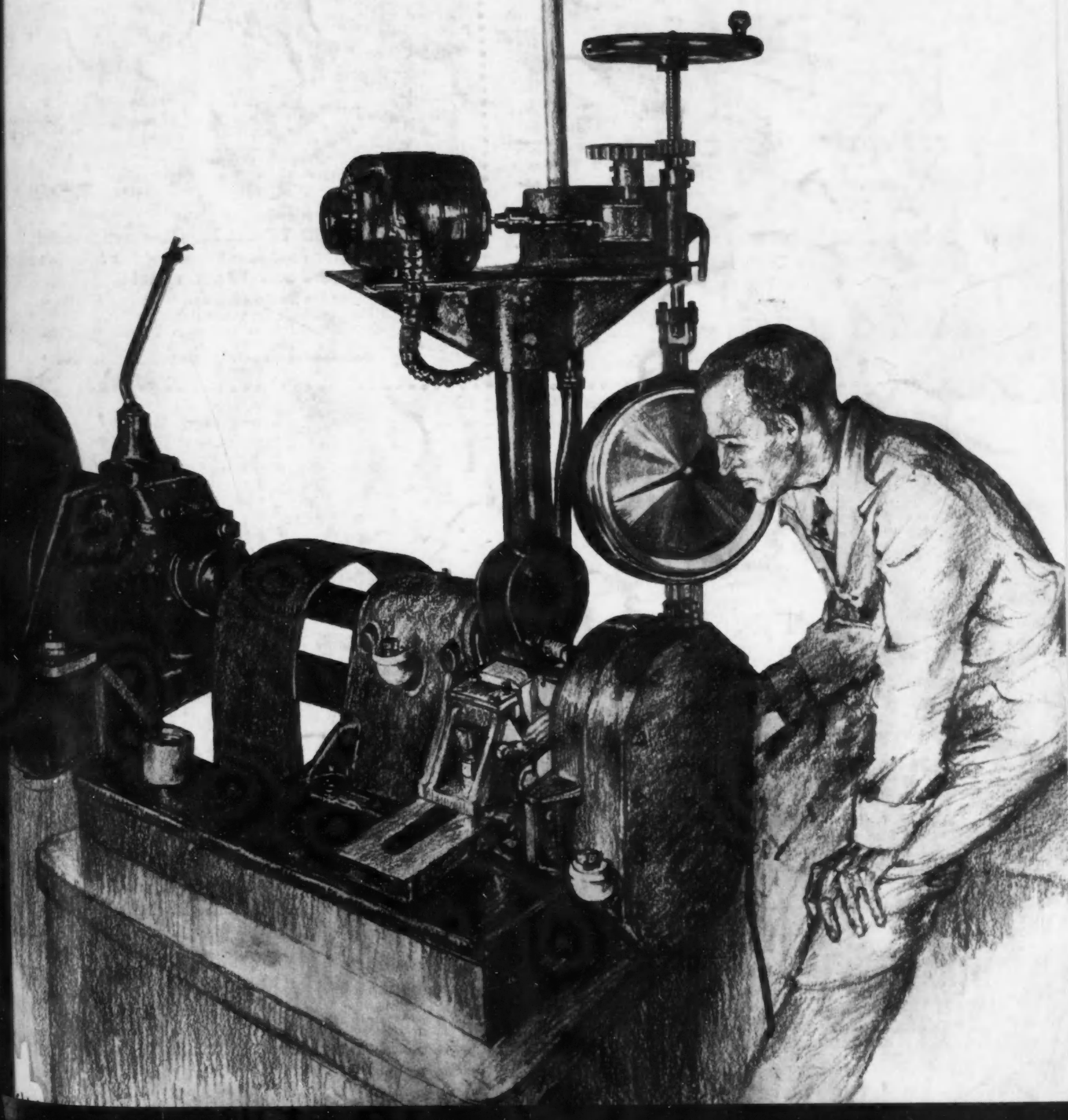
MAY 1949



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Technical Highlights Of Postwar Automobiles

(Part 1)

By Austin M. Wolf
AUTOMOTIVE CONSULTANT

(Part II, which covers the remainder of this paper, will be published in the June issue.)

PASSENGER-CAR production was virtually suspended during World War II and the stupendous wartime effort of the industry in the production of aircraft components and parts, tanks, military vehicles, munitions, and the like, is a matter of record and justifiable pride. The 1942 models, introduced in the Fall of 1941 but void of most chrome, were reintroduced in 1946 in unchanged form and design. In the closing years of the war there was considerable speculation regarding the postwar automobile, but in view of the insatiable demand for vehicles, production from available jigs, tools, and dies seemed more feasible than to attempt to make immediate modifications. No experimental work was permissible during the war years so that an immediate introduction of brand-new models was out of the question.

In the middle of 1946 Studebaker courageously pioneered a bold new design in which the powerplant and the passenger compartment were moved considerably forward, individual front fenders ceased to exist by being merged more completely into a front "square-ended" sheet-metal group balanced by a corresponding trunk mass at the rear; the detachable "rear fenders" projected a minimum amount from the widened body contour to break up an oth-

This paper was presented at the SAE National Passenger Car, Body, and Production Meeting, Detroit, March 8, 1949. (Multilithographed copies are available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

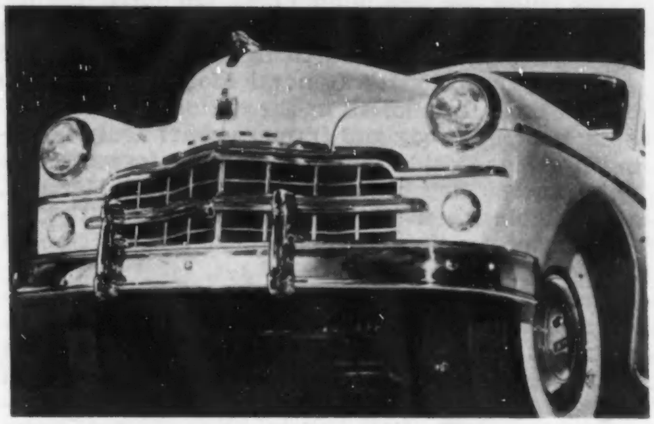


Fig. 1—Front view of Dodge

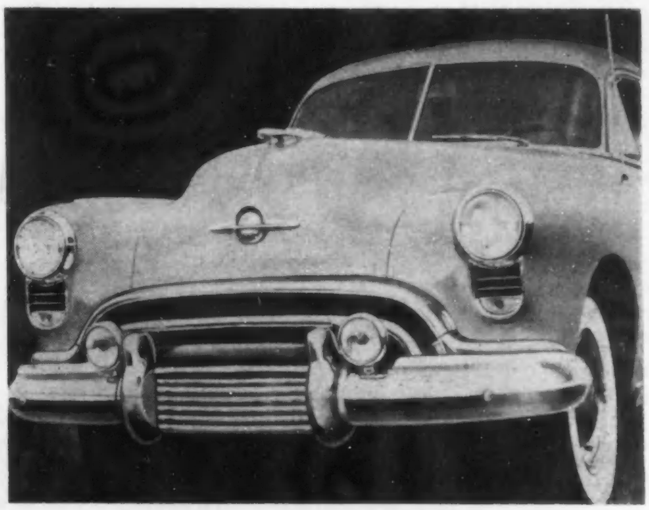


Fig. 2—Front view of Oldsmobile

Fig. 3—Front view of Studebaker

erwise monotonous panel running the length of the car. Kaiser-Frazer introduced in early 1947 their Darrin-designed, smooth-contoured cars with the flat front fender apron sweeping back along the body to join the rear fender, with a sharp corner at the belt line. Packard introduced its roadster in the summer of 1947, prior to the remaining models, in which large radius curves predominate to the exclusion of straight lines with emphasis on the body width below the well filleted belt. In early 1948 the Cadillac and the Futuramic Oldsmobile came upon the scene, the latter particularly forecasting the general conception of General Motors vehicles to come. Hudson followed in the spring with its low-hung monobilt body and frame, a rearwardly tapering low-placed side panel suggesting forward motion. Late spring unfolded the new Ford, Mercury, and Lincoln lines. Not until this year did the remainder of the industry get into full production on its new designs. (See Figs. 1-3.)

The American postwar cars incorporate the following features: low silhouette, the passenger compartment moved forward so that the rear seat is clear of the wheel housings, the glass area all around considerably augmented, rearward placement of narrower windshield posts, easier access and egress, interior dimensions increased, dual ducts leading to the body interior from the front grille for ventilation, and front and rear fenders that have either lost their identity or their individuality. Compression ratios have been upped in many cases and Cadillac has finally released the powerplant it has been working on for many years. With an eye to the future, the Kettering engine,[1] as found in the Oldsmobile, has a block design that will remain unchanged for a decade. (See Figs. 4 and 5.) The 1948 Buick dynaflow transmission marks a new performance in acceleration.

While there have been some slight increases in wheelbase, there is also a trend toward reducing it, as well as overall lengths. The wide space between the radiator core and the grille has been reduced, cutting down unnecessary overhang, as exemplified in the Chrysler-built cars. While the Pontiac has increased its wheelbase from 119 in. to 120, its overall length is 2 in. shorter. Plymouth has been reduced 6 in. on its P-18 in spite of the wheelbase going from 117½ in. to 118½. Dodge dropped only ⅞ in. but upped its wheelbase 4 in. to 123½ on the Coronet and Meadowbrook. DeSoto has a 4-in. greater wheelbase with only ¼ in. less overall, but with the front wheels cut to a greater angle to maintain the previous turning radius. Chevrolet has reduced its wheelbase from 116 in. to 115 and

[1] See SAE Quarterly Transactions, Vol. 1, October, 1947, pp. 669-679: "More Efficient Utilization of Fuels," by C. F. Kettering. An Oldsmobile equipped with a 6-cyl Kettering engine was shown at the meeting at which this paper was presented.

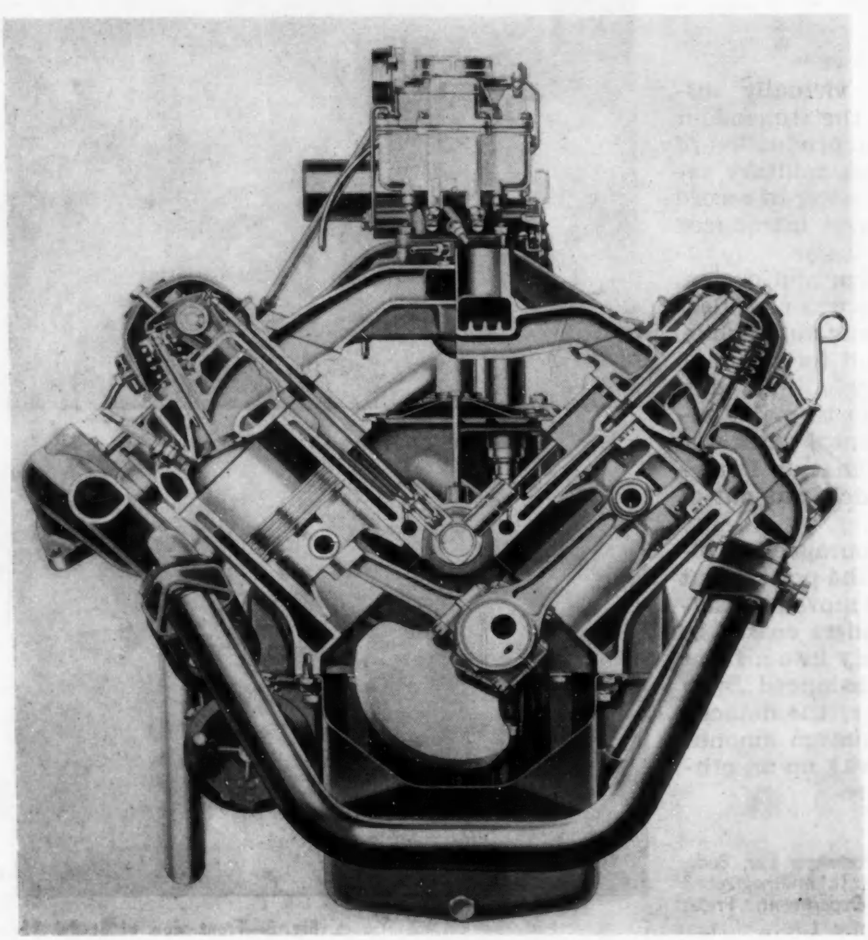


Fig. 4—Cross-section of Cadillac engine

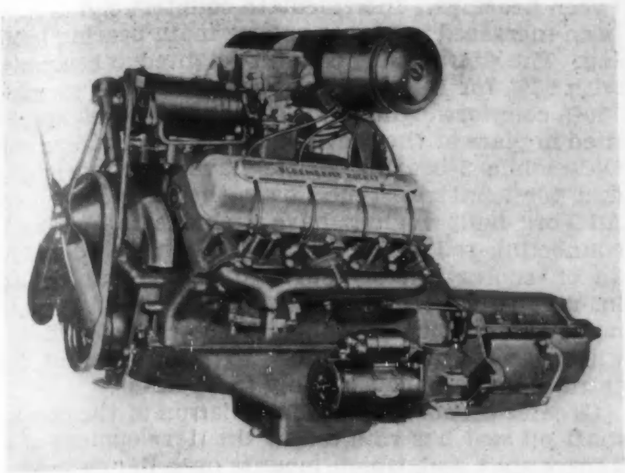


Fig. 5—Oldsmobile Rocket engine

the overall ¾ in. The Buick "50" and "70" have a 3-in. shorter wheelbase, 121 and 126 in., respectively. While the Lincoln has a 121-in. wheelbase, its companion, the Lincoln Cosmopolitan, has 125 in., the overall lengths being 212½ and 220 in., respectively. The trend toward shorter cars is indicated by the Plymouth P-17, with its 111-in. wheelbase and the Dodge Wayfarer with 115 in. Both of these shorter-wheelbase chassis are equipped with a 3-passenger coupe and a 2-door sedan. The Plymouth P-17 also has an all-steel suburban while the Dodge Wayfarer has a roadster.

There is a tendency to decrease weights. Chevrolet has painstakingly pared down wherever possible. The new Ford design is considerably lighter than the previous one.

A review will now be made of the various features in the new vehicles, and this will indicate the status of postwar designs as of this date. The details and trends hereinafter enumerated should establish a perspective on the subject. This study will indicate that the ultimate postwar car, with emphasis on initial cost and economy of operation, has not yet arrived.

Engines

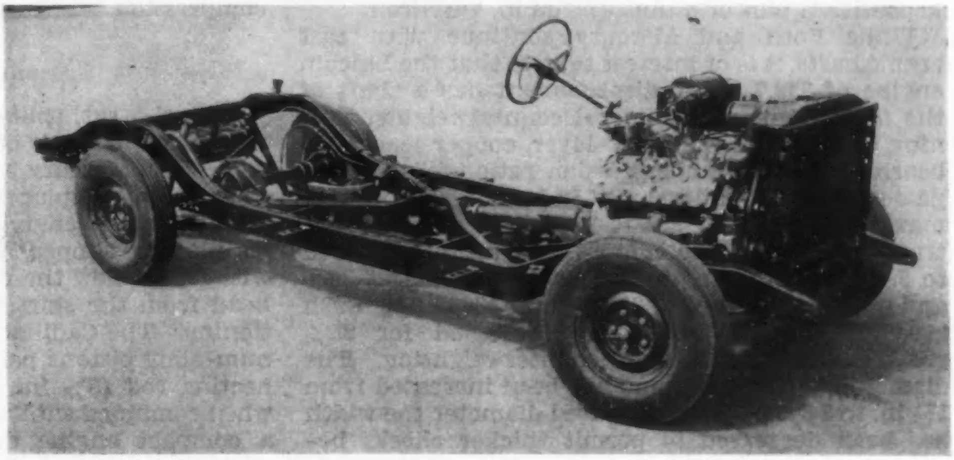
Compression ratios are generally on the increase, ranging from the incidental 6.5-1 to 6.6-1 in the Chevrolet, due to enlarging the size of the spark plug and changing from a concave to a flat inlet valve head, to 7.5-1 in the new Cadillac (6.70-1 for export). The Oldsmobile Rocket and the Chrysler 8 engines have a 7.25-1 ratio, the remainder of the Chrysler-built engines having 7-1. While the Cadillac and Oldsmobile engines will step up their compression ratios in the years to come, step by step, as better fuels become available, the test stand performánce of the Kettering engine at 12-1 indicates a 40% increase in mileage on 100-octane fuel, together with a jump in brake thermal efficiency of 32%.

There have been only four increases in engine size, namely the Mercury with its ¼-in. longer stroke, the Studebaker Commander with a ⅜-in. longer stroke and the redesigned Lincoln (Fig. 6) and Oldsmobile Rocket (Fig. 5) engines. The compression ratios in these four engines are 6.8-1, 6.5-1 (7-1 optional), 7-1, and 7.25-1, respectively. Inasmuch as the remainder of the industry has not increased displacements, it is believed that the peak has been reached. Cadillac has led the procession downward with a dropping of the displacement from 346 cu in. to 331. Packard has three engines, all with 3½-in. bore. One cylinder block serves two sizes. With strokes of 3¾, 4¼, and 4⅝ in., the displacements are 288, 327, and 356 cu in., and maximum bhp 130, 145, and 160, respectively. All have a 7-1 compression ratio. Pontiac's standard compression ratio is 6.5-1 but 7.5-1 heads are optional. Kaiser-Frazer is using 7.3-1. In 1948 Willys introduced a 6-cyl engine of 3 × 3½-in., 148.5 cu in. displacement, compression ratio 6.42-1, and developing 72 hp at 4000 rpm. The groundwork for the Crosley Cobra engine was laid in a Navy development for a lighter motor-generator set. The result was a 4-cyl water-cooled 44 cu in. engine, 2½ in. bore × 2¼-in. stroke, weighing 59 lb bare (131 lb with all accessories) and developing 26.5 bhp at 5400 rpm. The block consists of 125 parts made up of steel stampings and tubing, hydrogen-brazed into a unit, which mounts on a die-cast crankcase upper half. The avoidance of hot spots permits a compression ratio of 7.8-1.

Cylinders and Crankcase

A feature of the Cadillac and Oldsmobile designs is in the placement of the cylinder-head stud bosses

Fig. 6—Lincoln chassis



so that they are clear of the cylinder wall and are combined with the water-jacket wall instead (most of them being outside), preventing distortion of the bores, a feature introduced in the White Super-Power engine of 1939. The rigidity of the blocks, designed to withstand 1000-psi explosion pressure, ensures extremely smooth performance. The bell housings are cast integral.

High compression calls for improved cylinder-head gaskets. Victor Mfg. & Gasket Co. has worked out a design incorporating a core of approximately 1/32-in. thick asbestos millboard with a top layer of copper, which is formed into the water openings, and a bottom layer of steel formed into the combustion-chamber openings. The structure maintains the required strength of steel adjacent to the combustion chamber for resistance to burn-through or blow-out and the maximum corrosion resistance of copper in the waterway openings. The structure is a compressible gasket, used in the Oldsmobile V-8 but with steel throughout. Victor has also produced a dual-bead half-hard sheet-steel gasket structure for use in either valve-in-head engines or the L-head type in which the compressed thickness is 0.020 in. A rigid head and block are, of course, essential.

The new Cadillac has a 3 13/16-in. bore and 3⅝-in. stoke or a stroke-bore ratio of 0.951-1. Displacement is 331 cu in. This contributes to efficient operation by exposing about 12% less cylinder-wall area to flame and results in lower heat loss and greater cooling efficiency. The shorter piston stroke reduces frictional power losses by cutting down piston travel approximately 20%. At 4000 rpm it is at a rate of only 2400 fpm as against 3000 in previous models. The high mechanical and thermal efficiencies contribute materially to higher power output. The five main crankshaft bearings are placed in heavy bulkheads, which help to form a rigid, boxlike crankcase structure.

The Oldsmobile bore and stroke are $3\frac{3}{4} \times 3\ 7/16$ in., giving a stroke-bore ratio of 0.917-1 and a displacement of 303.73 cu in., compared with last year's Futuramic with 257.1, as well as being 10 in. shorter. The right bank is staggered ⅞ in. to the rear of the left cylinders. Fuel consumption is 0.533 lb per hp-hr and thermal efficiency 26%. Combustion chambers are not machined but care in casting plus machining the head face by locating from the cavity surfaces enables the individual volumes to be kept within plus or minus 0.06 cu in. variation.

While Ford and Mercury continue with cast crankshafts, it is of interest to note that the Lincoln engine of 336.7 cu in. displacement has a shaft of the forged type with integral counterweights, running in steel-backed, triple-layer, copper-lead alloy bearings. It has a compression ratio of 7-1 in the passenger car and develops 152 hp at 3600 rpm and maximum torque of 265 ft-lb at 2000 rpm. In the truck application the compression ratio is reduced to 6.4-1 with the engine developing 145 hp at 3600 and maximum torque of 255 ft-lb at 1800. The Nash "600" crankshaft has been redesigned for 25% greater stiffness and better counterweighting. The diameter of the crankpins has been increased from 1⅞ in. to 2 3/32. With increased diameter the width has been decreased to permit thicker cheeks between bearings. Intermediate counterweights have been increased in size to reduce main bearing loading. The shaft is 80% counterweighted as compared with 63% for 1948. The Ambassador shaft remains 100% counterweighted but eight counterweights are used in place of the previous four. The Cadillac and Oldsmobile 5-bearing shafts have greater rigidity and torsional resistance over the previous three. All Ford-built engines have a sludge trap in each connecting-rod pin consisting of the plugged end of an off-center radially outward passageway drilled in, and parallel to the axis of, the pin. Pontiac holds the balance tolerance of the crankshaft, flywheel, and clutch assembly to ½ in.-oz and the crankshaft balance assembly to ¼ in.-oz.

Greater economy in the installation of the crankshaft oil seal has resulted in the development of a 2-piece split seal, which mounts on a flange section of the block and outer flange of the rear main bearing cap. A supplementary seal is required for metal-to-metal surfaces on the flat of the lower main bearing cap and cylinder block, which is achieved by using a molded rubber section placed in a predetermined groove and abutting the split seal.

Flywheel timing marks have become less accessible due to high fenders and more accessories, so that most engines have their timing marks on the crankshaft damper. Houdaille Engineering Division has introduced a torsional vibration damper using silicone fluid in shear as a damping medium. The unit is being used by Packard, Lincoln, and Kaiser-Frazer, with Mack, Ford, Hercules, Cummins, and Reo in the heavy-duty field.

Higher compression ratios demand a new type of main and connecting-rod bearing material. The Buick, Cadillac, and Oldsmobile engines are using Durex 100-A with GM4167-M babbitt overlay. Cleveland Graphite Bronze Co. developed their F-77 material out of their war experience with aviation engines and in which they use a steel back together with a relatively high fatigue resistant bearing material of approximately 27% lead and the balance copper, with an 0.001-in. precision-plated surface layer of lead-base material, which is deposited as an alloy consisting of 10% tin, approximately 2% copper, and the balance lead. The thickness is controlled within 10% as deposited, thus eliminating the necessity for final machining. Unit pressures as high as 3600-psi projected area are satisfactorily handled and the material is now being used in high-compression test engines.

Piston and Rings

A steel-strut, split-skirt, oval, tin-plated, aluminum-alloy piston is being used by Mercury and Lincoln. The Mercury bore was increased from 3¾ in. to 4, giving a displacement of 255.4 cu in. The Oldsmobile Rocket engine is provided with steel-strut, tin-plated, slipper-type, aluminum-alloy pistons with slots below the ring belt, separating the piston head from the skirt to allow true crosshead functioning. The Cadillac T-slot, stanate-finish, aluminum-alloy pistons permit the use of a shorter connecting rod (6⅝ in. centers from 8¾ in.), which, when combined with a short piston stroke, results in a compact engine design of minimum size and

weight. The Cadillac and Oldsmobile pistons have a balancing pad below each wristpin boss. Top and bottom balancing pads on the connecting rods are automatically machined off so that small and large ends are reduced to the specified weights for proper balance and to the overall weights of 24.53 and 29.54 oz, respectively. The pistons, without rings, pin, or bushing, weigh 19.296 and 19.88 oz, respectively. Cadillac's 3 15/16-in. long piston has two 5/64-in. compression rings and one 3/16-in. oil ring; in this respect Oldsmobile differs only in having a 4-in. piston. The wristpin floats in both the rod and the piston. Rod centers are also 6⅝ in. as well as a crankpin journal 2¼ in. in diameter and 2 in. long. The wristpins are offset 1/16 in. in the pistons of both engines.

The Muskegon Piston Ring Co. is offering compression rings having a chrome flash on the cylinder contacting face for use in the piston top groove only in either the straight-face or torsional-twist type. The chrome is plated 0.002 to 0.004 in. for passenger cars and 0.004 to 0.006 in. for truck and bus engines. They have shown marked scuff resistance and reduced wear on both cylinders and rings. Most passenger-car engines are using first and second groove compression rings of either the torsional-twist or taper-face design. The trend is definitely toward narrower compression ring width, either 3/32 or 5/64 in., and of the thick-wall type. They provide better blowby control, reduce scuffing, and improve following ability. Plymouth is using two 3/32-in. compression rings, the upper being chrome plated. The "thread finish" of the base metal is duplicated in the plating process and lapped. Oil rings are generally wider in order to carry wide slots for maximum resistance to plugging and clogging. The cylinder contact lands are made very narrow in order to take advantage of high unit pressure for maximum oil-scraping ability. The entire oil-ring contour has been studied to give good streamlining to thwart plugging and wedge contour channels and radiused slots are often incorporated. The two lower Plymouth oil rings are 5/32 in. wide.

The Thompson steel-belted piston incorporates a graphite-coated, polished steel wire ring at the top of the skirt and just below the T-slot to control expansion. The graphite prevents surface adhesion between the piston aluminum alloy and the steel at the time of casting and affords a lubricant between them later when in operation. The Thompson U-flex piston ring has been used by Buick for several years for oil control.

The Nash "600" piston now carries four rings. The wristpin is now clamped in the connecting rod with bearings in each piston boss. Rifle drilling of the rods has been discontinued and the pin bearings are now lubricated by oil thrown off from the crankshaft and oil drained from the cylinder walls. The Ambassador engine retains rifle-drilling, using lengthwise holes to lubricate the wristpin.

The industry has in recent years become aware of the importance of safe and quick breaking of ring and cylinder surfaces. Bore finishes are more closely controlled, the limit being within the range of 15 to 25 micro-in. To mate rapidly, rings are coated on their periphery with tin, cadmium, or a chemical treatment such as Parco Lubrite. The importance of having a controlled thread depth and contour on the ring face is now appreciated because of its effect on scuffing and oil control.

The Koppers Co., Inc., has developed a high-tensile, centrifugally cast, heat-treated ring that, when spun, solidifies as white iron giving a tensile strength of 96,500 psi and a Rockwell C hardness of 26. The matrix is of the tempered martensitic type. The company is also porous chrome plating them. Sealed Power Corp. has found that the chrome-faced ring requires a compensating oil ring that has a higher unit pressure.

Valves

Breathing capacity has been increased by using relatively still smaller exhaust valve diameters as compared to the inlet. This is noted in the Cadillac and Oldsmobile engines, where ease of flow is still further increased by the inclined valve axes, which at the same time permit shorter rocker arms and reduce their inertia. The double-walled Cadillac valve cover plate insulates against noise and at the same time stabilizes temperature fluctuations, which will minimize the collection of condensation in a chilled engine. In view of the great interest in Kettering engines, particularly regarding valve materials, dimensions, spring pressures, and timing, data are submitted in Table 1, based on the AMA specification sheets. To eliminate sticking, the Oldsmobile cast-iron valve guides are Parco Lubrited, a step used by Packard for several years.

For heavy-duty service Eaton Mfg. Co. is producing Eatonite for valve and valve-seat inserts, an alloy whose hardness (approximately Rockwell C 40) is present when the material is applied and requires no further heat-treatment. It retains its hardness and is lead oxide resistant at elevated temperatures. Ford and Mercury now have solid valve guides, reducing the possibility of oil and air leakage at this point.

For long life of the exhaust valve and its seat, valve rotators are coming into use in the heavy-duty field. Rotation of the valve is caused by an inclined-plane principle in the Thompson rotator, whereas the offset mushroom tappet, with its inherent twist, imparts a movement to the valve during a spring-pressure-free period established by a built-in clearance of 0.002–0.005 in. in the valve-tip cup. Rotators are produced by Eaton and Thompson. The future economical passenger-car engine will assume a greater full-load capacity, especially with the transmissions to come, and greater cooling of the two intense hot spots of the engine—the exhaust valve and piston head center—will become a strict necessity. The hollow-head sodium-cooled valve of both these companies will help still further.

The Oldsmobile 5-bearing high-alloy camshaft (GM 1X cast iron) is Parco Lubrited. Its bushings are tin-plated on their exterior surface to guarantee full seating after they are pressed into position in the block. The push-rod seats and valve-stem pads on the rocker arms are induction hardened. The hydraulic valve lifters produced by GM Diesel Equipment Division are used by Cadillac, Oldsmobile, and Buick. The Oldsmobile lifter is an alloy casting with a diameter of 0.8424 in., plus 0.0003 minus 0.0000. Oil is fed from a galley and enters a

Table 1—Valve Data on Cadillac Engine (Left) and Oldsmobile Rocket Engine (Right)

Cadillac Engine

Intake Valve—
- Make Rich Mfg. Co.
- Material 3140 Steel
- Overall Length 4.539-4.559
- Actual Overall Diameter of Head 1.750
- Minimum Port Diameter 1 5/8
- Angle of Seat, deg 44
- Is Valve Seat an Insert? No
- Stem Diameter 11/32
- Stem to Guide Clearance 0.0005 to 0.0025
- Lift 0.330
  - With Valve Closed, lb 60 in. 1.696
  - With Valve Open, lb 135 in. 1.366
  - Length Out of Engine, in. 1.968

Exhaust Valve—
- Make Rich Mfg. Co.
- Material Head: N-82120 Stem: 8729
- Overall Length 4.539-4.559
- Actual Overall Diameter of Head 1.437
- Minimum Port Diameter 1 5/16
- Angle of Seat, deg 44
- Is Valve Seat an Insert? No
- Stem Diameter 11/32
- Stem to Guide Clearance 0.0015 to 0.0035
- Lift 0.330
- Spring Pressure and Length—
  - Outer—
    - With Valve Closed, lb 60 in. 1.696
    - With Valve Open, lb 135 in. 1.366
    - Length Out of Engine, in. 1.968

Operating Tappet Clearance (Hot or Cold)—Intake Automatic
Tappet Clearance for Valve Timing—Intake 0.001
Operating Tappet Clearance (Hot or Cold)—Exhaust Automatic
Tappet Clearance for Valve Timing—Exhaust 0.001
Hydraulic Valve Lifters? Yes

Valve Timing	At 0.001	Tappet Lift	
Intake Opens	19 deg	BUDC Piston Travel	0.121 in.
Intake Closes	83 deg	ALDC Piston Travel	1.365 in.
Exhaust Opens	53 deg	BLDC Piston Travel	3.085 in.
Exhaust Closes	49 deg	AUDC Piston Travel	0.778 in.

Valve Timing Marks (Flywheel, Vibration Damper, None) None

Oldsmobile Rocket Engine

Intake Valve—
- Make Various
- Material SAE 3140 Steel
- Overall Length 4.855
- Actual Overall Diameter of Head 1 3/4
- Minimum Port Diameter 1 7/16
- Angle of Seat, deg 45
- Is Valve Seat an Insert? No
- Stem Diameter 0.3425-0.3417
- Stem to Guide Clearance 0.00175 to 0.00355
- Lift 0.222
  - With Valve Closed, lb 65 in. 1.777
  - With Valve Open, lb 140 in. 1.447
  - Length Out of Engine, in. 2 3/32

Exhaust Valve—
- Make Various
- Material Silchrome XCR Steel
- Overall Length 4.8435
- Actual Overall Diameter of Head 1.469
- Minimum Port Diameter 1 1/4
- Angle of Seat, deg 45
- Is Valve Seat an Insert? No
- Stem Diameter 0.3938-0.3930
- Stem to Guide Clearance 0.00225 to 0.00405
- Lift 0.222
- Spring Pressure and Length—
  - Outer—
    - With Valve Closed, lb 65 in. 1.777
    - With Valve Open, lb 140 in. 1.447
    - Length Out of Engine, in. 2 3/32

Operating Tappet Clearance (Hot or Cold)—Intake None
Tappet Clearance for Valve Timing—Intake None
Operating Tappet Clearance (Hot or Cold)—Exhaust None
Tappet Clearance for Valve Timing—Exhaust None
Hydraulic Valve Lifters? Yes

Valve Timing—			
Intake Opens	14 deg	BUDC Piston Travel	0.060 in.
Intake Closes	50 deg	ALDC Piston Travel	2.94 in.
Exhaust Opens	50 deg	BLDC Piston Travel	2.97 in.
Exhaust Closes	14 deg	AUDC Piston Travel	0.068 in.

Valve Timing Marks (Flywheel, Vibration Damper, None) None

drilled hole at the side. In the process of selective matching of the plunger and body, the diammetral clearance is held within 0.00016–0.00027 in. In the cast-iron body lifter for Buick the diammetral clearance is held within 0.0005–0.0007 in. The outside diameter is 0.9985 in., plus 0.0000, minus 0.0010. In the latter, oil from the rocker arm shaft is fed into the top of the tappets via drilled passageways in the rocker arm and clearance adjusting stud into the push-rod, the lower end of which has a hole mating with a corresponding one in the top of the lifter plunger. The push-rod has a drilled hole near the top to prevent air lock after starting and to prevent a pressure buildup in the rocker arm shaft. The hole also feeds oil down to the outer surface of the lifter to lubricate it in its crankcase boss. A shroud on the upper end of the push-rod covers the hole and deflects the oil downward so that it will not be thrown up on the inlet valve. New heavier inner and outer valve springs are used to offset the spring in the lifter and the increased inertia load produced by the heavier lifter and the column of oil in the push-rod, which is made of thicker material to offset the loss of column strength produced by the drilled hole. The acceleration of any engine with a transmission of the dynaflow type would raise the noise level with ordinary tappets and the hydraulic type eliminates it. The Nash Ambassador camshaft bearings have been reduced from 6 to 4, providing increased oil pressure when idling due to fewer points for leakage.

Lubricating System

Chevrolet has replaced gravity-feed timing gear lubrication by pressure feed. A passage from the front camshaft bearing leads oil between the engine front plate and the cylinder block. From that point the oil is directed into the timing gears by a

Table 2—Parco-Lubrizing Operation on Cylinder Bores

Dodge Sequence of Operations

1. Cleaning at 190 F—1 3/4 min—spray.
2. Water rinse at 200 F—1 3/4 min—spray.
3. Cleaning at 150 F—1 3/4 min—spray.
4. Air blowoff.
5. Parco Lubrite at 200 F—15 min—solution pumped into bores only.
6. Water rinse—10 sec—room temperature—spray in bore areas only.
7. Lap—cast-iron lap plus oil—6 spindle.

Blocks enter the Parco-Lubrizing operation on an overhead conveyor, which carries them through the 3-stage washer. They are then removed from the conveyor and, by means of an overhead hoist, placed head down on a series of stationary jigs, or fixtures, which overhang the Parco-Lubrite tank. Solution is picked up from the tank by individual pumps and carried up to the fixtures, which are arranged so as to permit entry of the solution to the cylinder bores only. By means of a standpipe in each bore, the solution is overflowed in a continuous stream and returned to the Parco-Lubrite tank.

After Parco-Lubrizing, the blocks are passed on to a final lapping operation to condition the cylinder bores for fitting of pistons and rings. At the Dodge main plant this is accomplished by means of a cast-iron lap with oil. A vertical 6-spindle machine locates the block properly and four strokes are taken simultaneously in each bore.

Chrysler-Jefferson Sequence of Operations

1. Alkali clean at 170 F—spray.
2. Water rinse at 170 F—spray.
3. Water rinse plus conditioner—170 F—spray.
4. Air blowoff.
5. Parco Lubrite No. 2—205 to 208 F—immersion—entire block.
6. Water rinse at 170 F—spray.
7. Air blowoff.
8. Burnish with steel brush.

Unlike the Dodge operation, this process is entirely automatic. Blocks are placed on an automatic conveyor and carried through the above sequence of operations without removal or transfer. In addition, this installation provides for Parco Lubrizing the entire block, eliminating the necessity for fixtures to confine the solution to cylinder-bore areas.

After passing through the various spray cleaning and rinsing operations, the conveyor descends to allow immersion of the entire block in the Parco-Lubrite solution. The Parco-Lubrite tank is of sufficient length to allow for adequate processing time at a fixed conveyor speed.

The final finishing operation at the Chrysler-Jefferson plant differs, too, in that steel brushes are used to burnish the cylinder bores. This is accomplished in much the same way and all bores are burnished simultaneously.

nozzle. The engine oil filler pipe has been eliminated and replaced by a filler opening on the top and front end of the valve cover plate, where it is accessible from the front rather than from the side. The dip stick has been lengthened.

In the Oldsmobile Rocket engine the pressure of the oil in the rocker arm shafts is controlled by an intermittent alignment of two holes, 120 deg apart, in the cam journals. Two of the camshaft bearings have drilled holes to supply and meter this oil to the two rocker shafts. Oil release discharge carried below the surface of the sump level to prevent aeration was adopted on the Nash Ambassador engine in 1948.

A new low-restriction type of oil filter element is now used by Buick in the AC type B-10 filter. The oil supply for the overhead system now comes through the oil filter and ensures clean oil for the rocker arms and valve mechanism. Since the direct line to the overhead is eliminated, there is more oil available for the bearings at low engine speeds. To ensure a supply of oil to the overhead system at all times, a relief valve is provided that will open at 7 to 9 psi if the filter element is not changed before the restriction rises to that pressure. Another safety feature of the new installation is that the cover of the filter will not seat if the wrong type of element is installed. Changes are recommended every 5000 miles after the initial 2000. The Oldsmobile Rocket engine is provided with a full-flow filter, the AC type PM-9. As with other AC filters of this type, it is composed of an accordion-shaped filter element made of resin-treated paper, relatively porous but with the pores sufficiently fine to accomplish maximum filtering with low pressure drop. A bypass valve opens at a 5- to 7-psi pressure differential and permits direct flow of oil from the pump to the bearings. The unit is installed on the engine so that there are no external pipes and the readily replaced element is recommended for change at approximately 5000 miles. Packard was the first to discard the bypass method of feeding the oil filter in 1940. Chrysler followed suit in 1947. The AC shunt type of paper filter is made of a heavier and denser paper; it is also formed in an accordion-pleated element.

DeLuxe Products Corp. has designed a removable sump for its filters so that, in changing the cartridge, it is not necessary to drain sludge and water from the filter sump. Purolator Products, Inc., is producing its micronic filter with plastic-impregnated paper in which the pores can be kept as small as 0.00004 in. The accordion-pleated paper, if stretched out, reaches up to 10-ft in length.

The schedule for engine break-in announced last fall by Dodge called for an operating speed of 40 mph for the first 250 miles, thereafter 5 mph being added for each additional 25 miles traveled until at 350 miles the engine was considered run in. This was made possible by Parco Lubrite, which is a treatment of properly cleaned iron (and steel) articles resulting in the chemical conversion of their surfaces to a nonmetallic, oil-absorptive coating consisting chiefly of iron and manganese phosphate. Since the coating results from actual reaction of the solution with the metal surface, it becomes chemically combined with the base metal and has greater adherence than could be expected from any coating depending upon natural attraction. The process results in the preferential attack of metal burrs, resulting from machining or abrasive operations, by the solution due to their increased unit area whereby they are principally dissolved and removed during the coating formation. The procedures employed at the Dodge and Chrysler plants are detailed in Table 2.

Fuel System

The lowered hoods have resulted in considerable emphasis being placed on the reduction in the overall height of carburetors by their manufacturers. Carter Carburetor Corp. has developed a new downdraft dual carburetor, appreciably shorter and embodying their built-in antipercolating system. Eclipse of Bendix Aviation Corp. has also been lowering its carburetors for Buick, Dodge, and Studebaker Champion engines. Improvements include an electrically controlled dashpot as well as an airflow operated switch for modifying the overdrive kickdown operation. Power line and transmission developments, such as the fluid flywheel, overdrive, torque convertors, and the various semiautomatic transmissions, call for different carburetor performance requirements over the conventional clutch and hand shifting, and which all carburetor manufacturers have made. The Oldsmobile carburetor made by Rochester Products Division, GMC, has a horizontal air inlet permitting a low disposition of the air cleaner. The temperature of the float bowl is reduced since it is suspended from the top cover of the carburetor housing, thus establishing a long heat-flow path and an intervening gasket. Besides, it is brushed by the incoming air. A vacuum-operated power jet consists of a neoprene diaphragm, stem, and calibrated spring located in the cover to provide the richer mixture needed for wide-open throttle operation. The concentric float bowl ensures a constant fuel feed regardless of acceleration, deceleration, or road angle.

Better distribution is always obtained when the intake manifold is parallel to the ground in spite of the engine crankshaft inclination on the chassis. This expedient is now incorporated in Ford-built engines. The Holley carburetor used on the Mercury and Lincoln isolate and cool the float bowl in the same manner as on the Oldsmobile by suspending it in the air stream between the air cleaner and the carburetor inlet. There is a clean air intake for the automatic choke heat control by piping filtered air from the carburetor air horn through a heat tube pressed into the manifold hot spot. This prevents dirt deposits from interfering with proper heat transfer and correct action of the mechanism. AC has introduced two types of lowered air cleaners and silencers to take care of the restricted conditions. The first is a concentric type in which the silencing element, instead of being either above or below the oil bath cleaner, now surrounds it, permitting a much lower design but of considerably larger diameter. The other type is the crosswise horizontal unit used on the Oldsmobile Rocket engine, being much longer than the ordinary silencer and with the oil bath cleaner attached at one end.

Plymouth and Mercury now include the automatic choke as standard. The Chevrolet hand choke, when operated, establishes a fast idle through linkage at the carburetor. The accelerator pump piston is entirely submerged in gasoline under all operating conditions, the fuel acting as a lubricant and assuring uniform operation of the pump at all times. The leather is constantly wet and maintains a continuous seal, whereas in its elevated position in the old design, it would dry out. The air cleaner is 3½ in. larger in diameter and almost 2½ in. shorter; a slightly greater volume is provided in the silencing chambers and, by redistributing this larger volume, more effective silencing is obtained. The filter element is larger in diameter and contains 1¼ oz more material.

On the Buick series "70" cars better results have been obtained by lowering the cleaning element 3/16 in. closer to the oil surface with further improvements in baffling to prevent oil pullover into the silencing body. A new large baffle has been put in the oil bath instead of on the cleaner element with a smaller baffle replacing the suction and antipullover baffle that was formerly located on the cleaner element. The silencer is of the single-chamber type with a silencing chamber increased ¾ in. in length. The series "40" and "50" air cleaners and silencers have been retained with an improvement in tuning.

On many of the newer cars the throttle cracker linkage has been deemed unnecessary and so has been removed because the starter button has been placed on the instrument board, leaving the right foot free to rest on the accelerator pedal. A temporary fast idle is desirable however, to prevent stalling when the gears are shifted and the engine is cold. In spite of its incorporation, Chevrolet holds on to the hand throttle knob. The combined fuel pump and vacuum booster for the windshield wiper is now available on Plymouth as special equipment. It also retains the Oilite fuel filter. Chevrolet and Plymouth continue to strap the fuel tank to the floor of the underbody. In the Pontiac tank, braces have been soldered to the tank assembly to the right and left of center, holding the floor and ceiling of the tank to a uniform dimension. On the left side the filler pipe is extended to fit snugly into aligned holes in the tank side and the left brace. A vent hole in the top of the filler pipe just inside the tank provides a better escape for air displaced in the tank. Chevrolet has also shifted the filler to the left side and added a vent tube to the filler neck, permitting a 70% increased filling rate. The tube is attached to the outside of the filler pipe for its full length and extends from an embossment on the tank above the filler pipe to a point where

it enters the pipe less than 2 in. below the filler cap. An electromagnetic fuel gage is used on all Chrysler-built cars, whereby the previous radio static suppressor is now eliminated, since the new gage has no contact points.

Two methods have been brought out whereby the continuous use of premium fuel can be avoided. In the Thompson Vitameter,[2] an alcohol-base solution is fed into the intake manifold during periods of low manifold vacuum, suppressing knock that would otherwise occur. In the Holaday two-fuel carburetor,[3] premium fuel is only fed under like depression by means of a vacuum-controlled solenoid selector of the two needle valves and at all times from the accelerator pump.

Exhaust System

The Nash exhaust pipe now runs around the forward end of the engine to avoid the possibility of overheating the equipment on the left-hand side, such as the generator, starter, steering gear, water pump, and battery. Chevrolet provides a sealed unit system in which all joints of the exhaust pipe flange, the exhaust pipe, and the muffler are electrically welded together, making a unit structure and providing greater protection against leakage. The muffler is equipped with four silencing chambers instead of three. If the muffler must be replaced at any time, the exhaust pipe is sawed through to remove it and the replacement muffler attached with clamps. In the Cadillac and the Oldsmobile Rocket engines the exhaust from the left-bank manifold crosses over to the right under the shallow portion of the sump. The common exit for the two banks is at the rear of the right manifold. In the Ford-built engines the discharge from the left-bank manifold is carried forward around the front of the engine and enters the front of the right-bank manifold with the common exhaust at the rear thereof in lieu of the previous under-engine crossover.

Cooling System

A chevron seal is used on both Nash models enclosed in a metal cartridge, installed and serviced as a unit. It is mounted and sealed in the pump body by means of a rubber O-ring, permitting the entire seal and bearing cartridge to align itself to the shaft and yet absorb radial loads imposed by the hydraulic forces on the impeller. The pump housings are of aluminum alloy. In the Mercury and Lincoln blocks the water flow has been rerouted from the radiator to the rear of the block where it enters the head in volume. Pontiac has lowered its water pump in view of the lower hood and radiator grille opening. The fan is now piloted directly on the water pump shaft instead of the water pump pulley flange. The fan-to-crankshaft drive ratio is 1-0.88. This provides a slight reduction in speed, which, together with less pitch, provides quieter operation. The fan diameter was increased from 18 in. to 19.

More manufacturers are taking advantage of pressurized cooling to reduce the size of the radiator and give improved cooling under heavy engine output. Twelve to fifteen-psi caps are being used (Buick dynaflows under arduous conditions) that permit maximum temperatures of 240-250F. Settings of 3 to 7 psi are common. Water or other coolant loss is prevented and satisfactory performance of the car is materially assisted, even when climbing high or steep hills, as well as in high altitudes. In order to prevent the possible installation of a higher pressure cap on a radiator designed to operate with lower pressures, AC has standardized their caps together with the filler necks in three sizes. The lower pressure cap can be installed in the higher pressure neck, but the reverse cannot be done. This extends the possible range of servicing without the possibility of damage from radiator rupture, provided it is not designed for the higher pressure. An important advantage of pressurized cooling is the prevention of cavitation by providing pressure on the inlet side of the pump. The bellows type of thermostat is based on a cooling system under atmospheric pressure. In the sealed system the vapor pressure counteracts the motivating force of the vapor pressure within the bellows and prevents proper action of the thermostat. The Dole Valve Co.'s metallic type of thermostat overcomes this action, as well as its new capsule arrangement containing a special powder that expands on heating. The latter is standard equipment on Nash.

A smaller, lighter radiator is made possible with the new Cadillac engine because friction and heat losses have been reduced to a minimum. The new cooling system requires only 18 qt of coolant compared to the previous 25. A new integral casting comprising water pump housing and inlet and outlet water manifolds eliminates all hose connections in the Cadillac engine except those running to the lower and upper radiator tanks. The unit is supported on the front end of the block by the right and left inlet flanges into the jacket of each bank and on the outlet flange of each head. In the Oldsmobile Rocket engine the water pump, its inlet, and the oil filler pipe boss are an integral part of the front timing cover casting.

The Chevrolet radiator is shorter and wider, due to the lowered hood, the frontal area is increased more than 8% to 408 sq in. and the cooling capacity is 16 qt instead of 15. The fan is closer to the upper edge of the core and sweeps a greater portion of the high-temperature area. The air cell size is now 0.22×0.560 in. instead of the previous 0.20×0.560. Five added rows of water passages have been incorporated, due to the increased width, and the lowered height permits a greater flow of coolant at high speeds. The radiator is tested at a pressure of 10 psi and the filler neck permits the use of pressure caps operating at either 7½ or 4 psi. A curved radiator hose is now used on the Chrysler-built cars to reduce the strain on the radiator top connection; a brass-tube radiator top tank fitting and a distrib-

[2] See SAE Transactions, Vol. 53, (June) 1945, pp. 358-372: "Alcohol-Water Injection," by A. T. Colwell, R. E. Cummings, and D. E. Anderson.

[3] See "Progress Report on Dual-Fuel System," by W. M. Holaday. Digested in **SAE Journal**, Vol. 56, October, 1948, pp. 23-24 under title, "Feeding Two Fuels Brings Octane Thrift."

uting baffle in the top tank are incorporated. The very promising Eaton dynamic fan drive consists of a freely rotating fan under the influence of an eddy-current clutch energized by a thermostatic control element located in the lower radiator hose.

Ignition System

All ignition coils are now mounted on the engine close to the distributor, minimizing disturbance to radio and television reception, thanks to the work of the joint SAE-RMA committee on ignition interference.[4] With a reduction of 75% in the length of high-tension leads, Pontiac has found by tests that the terminal voltage at the spark plugs has increased 8%; they adopted the suppressor in the distributor cap last year. This year all Chrysler-built cars have a 10,000-ohm suppressor so located, and combined with a low-resistance soft carbon tip acting as the center conducting brush in the distributor cap, eliminating the previous condenser between the coil and the distributor to prevent interference. Spark plugs with a 10,000-ohm resistor[5] and a wider gap setting are now standard equipment, improving engine performance during idling and low speed and minimizing spark gap growth. The bakelite distributor cap has an exterior surface free of notches or index slots that might allow water to enter. It has two concentric ring barriers around the raised center carbon tip boss and one radial barrier. Adequate ventilation and drainage are provided by a ¼-in. hole in the distributor base. A barrier in the casting protects this hole against direct entry of water. All ignition cable is neoprene jacketed. To provide greater theft protection, an armored ignition cable is used at the rear of the ignition-starter switch through to the engine compartment.

Chevrolet has switched from 10- to 14-mm spark plugs, using the same type of insulator but with a larger shell, whereby the installation torque thereon is less critical. In addition, the large clearance volume in the combustion pocket of the 14-mm shell makes the possibility of fouled plugs more remote. Chevrolet has given up the polarity-reversing switch to permit the adoption of a simplified distributor and the substitution of a direct coil-to-distributor wire in place of the reversing switch harness. A condensor with a capacity of 0.2 microfarads replaces an 0.3 unit to get the necessary protection against pitted points. For better sealing against the entrance of moisture into the distributor, vinylite nipples replace the former rubber, being far more durable.

The Oldsmobile Rocket distributor has two concentric ring barriers. Mercury employs a single breaker arm in place of the two formerly used. All Ford-built engines now have accessible distributors, gear driven from the camshaft. Oil replaces wax in all Delco-Remy coils. The Coralox insulator in AC spark plugs provides high heat conductivity, greater resistance to deposits of lead compounds, greater strength, better electrical insulating properties throughout the operating temperatures, and better bonding.

Holley Carburetor Co. and the Ford Motor Co. worked out cooperatively the vacuum advance control of the Ford and Mercury, which is considered the equal of a combined vacuum and centrifugal control. The carburetor is drilled with two interconnected pressure take-off holes, one at the throat of the main venturi and the other adjacent to the edge of the throttle plate. A small tube connects these holes with the distributor diaphragm mechanism. The selective size and location of these holes govern the characteristics of the pressure groove. The breaker plate returns to the fully retarded position at once when the throttle is closed to idling position. The distributor shaft is one piece from the drive gear to the cam, eliminating backlash.

Starting and Lighting

A snap-lid metal cover is placed above the Pontiac and Oldsmobile batteries, providing protection for the service man when working on the car and shielding the battery from water, dirt, and foreign matter from above. The Pontiac conical battery filler neck gives visual evidence when looking down through the opening to the electrolite level, which is correct when it is touching the base of the cone. A slot is molded into the side wall of each neck to provide a vent. Chevrolet has moved its battery to a bracket on the right front fender skirt.

Generator outputs have been stepped up to 35 amp to take care of extra electrical equipment. The Cadillac generator is located out and alongside the front right valve cover. A triangular cog-built drive takes in the generator, fan water pump, and the crankshaft. The generator of the Oldsmobile Rocket engine is centrally located forward and above the V block and is belt driven from the rear groove of the fan water pump pulley. The forward groove takes the crankshaft V belt. Leece-Neville Co. is enjoying considerable popularity with its police car alternator, using a rectifier to change over to d-c current, thus avoiding commutator interference. The Plymouth starter has an eccentric on the drive barrel to ensure engagement at extremely low temperatures. Greater starting dependability is aided by heavy silver contacts in the starting switch. All Chrysler-built cars have a combination ignition-starter switch. The ignition key is turned and held beyond the ignition-on position. As soon as the key is released, a spring automatically returns it to the ignition-on position. Pushbutton starting on the instrument board solenoid switch has been adopted by Chevrolet, which found that removal of the chamfer on the ring gear actually decreased gear-tooth wear. The chamfer on the starting pinon is retained. Pontiac has also adopted the pushbutton. On the Oldsmobile with the hydramatic drive, a neutral safety switch is installed,

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[4] See "Automotive Industry's Participation in Reduction of Radio and Television Interference," by P. J. Kent. Digested in **SAE Journal**, Vol. 57, March, 1949, pp. 17-19 under title, "Television Brings Change in Car Ignition." Paper based on work done by SAE Vehicle Radio Interference Subcommittee. Full report of subcommittee not published, but occasional stories on the work of the subcommittee have appeared in the "Technical Committee Progress" section of **SAE Journal**.

[5] See "Modern Approach to Ignition," by C. Cipriani and L. H. Middleton. Digested in **SAE Journal**, Vol. 56, October, 1948, pp. 47-50, 57 under title "Ignition Theory Undercovers Electrical Design Factors."

which makes it impossible to operate the starter unless the selector lever is in neutral. Nash continues with clutch pedal starting.

Dodge headlights are located 3¼ in. higher and 5¾ in. farther apart; Plymouth 1⅜ and 4 in. A bull's-eye lens has been placed in the center of each headlight lens on the Chrysler-built cars to intensify the center of the beam. This particular lens is used on a metal-backed sealed-beam bulb unit manufactured by the Electric Autolite Lamp Division. The average 1949 car has 18 bulbs and one model has 32. If a car were to be equipped with all the bulbs presently used, it would have a total of 46 and the connected load would add up to 83 amp, as indicated in Table 3. There is never an occasion when all the bulbs would be used at once, hence the 35-amp generators of the new cars can handle the usual night time lighting load with ample reserve for ignition, heater, and radio, with a surplus for charging the battery.

Since the war there has been a general trend toward the return of the so-called hard-rubber battery container in place of the composition type used when there was a scarcity of rubber. Even with the improvements that have occurred in the composition container, the rubber type is still superior in its ability to withstand mechanical and thermal shocks. Every battery company has made efforts to beautify the "black box," and we now see some batteries with plastic, rather than rubber, vent plugs.

Table 3—Lamps Used on Postwar Car

Position	Number	Current, amp Connected	Current, amp Used While Driving
Headlamps	2	15-11	*
Fog lamps	2	11	
Parking	2	1.2	
Front direction signals	2	5.6	
Ornamental lamps	1	0.6	
Spotlamps	1	5.0	
Engine compartment lamps	1	1.9	
Trouble or utility lamps	1	3.9	
Instrument panel lamps	4	2.4	*
Headlamp beam indicator	1	0.2	*
Direction signal indicator	2	0.4	
Parking brake indicator	1	0.4	
Backup indicator	1	0.2	
Other indication	1	0.2	
Clock lamps	2	0.8	*
Glove compartment	1	0.4	
Radio Dial	2	0.8	*
Ignition Switch	1	0.4	*
Map lamp	1	1.9	
Driver's compartment lamp	1	1.9	
Interior—Dome	1	1.9	
Interior—Side	2	1.2	
Step lamps	4	7.6	
Tail lamps	2	1.2	*
Rear direction signal	2	5.6	
Stop lamps	1	2.8	
License plate lamps	1	0.6	*
Backup lamps	2	7.8	
Rear compartment or truck lamps	1	0.6	
Total	46	83.5	21.4

Engine Mountings

An integral extension on each Mercury and Lincoln water pump housing forms a front engine support. The Ford and Mercury rubber mountings are both lighter and more resilient than the previous type, which was made possible by the discarding of the torque-tube drive.

The former Chevrolet front mountings permitted freedom of movement in all directions and in the new design restriction is obtained in a horizontal plane by the use of an upper cup member, the sides of which fit down over the lower plate, which has edges slightly upturned. The intervening space is filled with rubber except for a slight peripheral cavity that avoids complete restriction of normal horizontal and vertical engine motions but limits the movement to a predetermined maximum. Tests indicated that, when the clutch was engaged at certain low speeds, vibrations of seemingly negligible amplitude in the clutch would be synchronized with horizontal vibrations of the same frequency occurring in the engine. The total intensity was great enough to be felt by the passengers and the new front mountings were especially designed to snub this buildup.

Clutch

Since 1944 Borg & Beck Division has produced its type E strap drive clutch. Ordinarily, the pressure plate of a clutch is provided with milled lugs that project through the cover stamping to hold the two concentric and in driving relation. Rubbing between the lug and the edge of the lanced hole introduces a certain amount of friction and sometimes causes squeak. It is a difficult point to lubricate because of the heat and centrifugal force that are developed in normal operation. Also, particularly in the large clutch, the clearance between the lanced hole and the lug is sufficient to allow a small amount of shift in the pressure-plate position, thus affecting accuracy of balance. By the use of four pairs of tempered steel straps riveted at one end to the cover stamping and bolted at the other to the pressure plate, the two parts are held in fixed concentric relation and there is no possibility of friction squeak when the clutch is released.

The Borg & Beck driven plate damper assembly utilizes a Belleville washer, which is flat in the assembled position at the rear of the driven plate hub, transferring its load through a pressure ring to a group of three shims or spacers on the rear side of the hub flange. Another group is assembled on the opposite side and, when the hub oscillates, it wears on the spacing washers over six surfaces, which tends to give a more uniform friction and reduces wear to a negligible point. Frictional control of the driven plate hub is old but the use of the Belleville washer to obtain the desired load is a relatively recent development. The washer is particularly effective because it has a very low deflection rate at or near the flat position. Consequently, considerable change in the dimensions of the associated parts will not appreciably change the pressure in the friction washers. It is possible to hold friction limits to a much lower range than can be accom-

plished when the side plates alone are deflected to obtain the desired load. When different loads are required, different Belleville washers are used, which, in combination with various damper springs, makes it practical to vary the damper characteristics over a broad range.

Long Mfg. Division has also incorporated the Belleville washer in a number of its units. It has added its models 12CF and 13CF to its line since the war, these being of the semicentrifugal type with a well-ventilated pressure plate and antifriction rollers in the lever system. All Ford-built cars have Long clutches except the Mercury, which is Borg & Beck equipped. Cadillac uses the Long clutch on cars equipped with the conventional transmission; Packard clutches incorporate the Torbend design of driven member. All of the Pontiac driven members and a portion of the Buick are of the Torbend disc design, these two companies using pressure-plate assemblies made by Inland Mfg. Division, GMC.

To obtain a more rigid mounting with reduced friction, the Plymouth clutch pedal shaft is mounted through the frame on needle bearings. The wider bodies on all cars have resulted in the clutch pedal and all the other driver controls being moved farther to the left. Several manufacturers insert a fabric strip in the clutch beam linkage to prevent transmission of engine noise to the body. To eliminate rattle, Pontiac has provided a new inner clutch countershaft bearing consisting of a steel cup, which supports a rubber insulated brass bushing. Sufficient space has been provided in the cup to hold a lifetime supply of viscous chassis lubricant. The rubber insulation breaks the metal-to-metal connection between the powerplant and the clutch pedal. Nash provides an aluminum clutch housing, saving 12½ lb in weight over cast iron.

Transmission

Any modifications in the conventional transmission, as well as overdrive units, are minor and do not incorporate any engineering changes. More rigid connecting links and new antirattle springs are used by Plymouth. Movement of the gear shift lever on rough roads and under acceleration has been reduced through the location of the bellcrank on the frame instead of on the engine. The speedometer take-off is on the left side of the transmission to protect the cable lubricant from exhaust heat. Chevrolet has dropped the vacuum shift. The shifter lever is longer and the upper portion of the control shaft is no longer enclosed in a housing. Both rotational and pivotal movements are transmitted to the gear shift upper control shaft, which is attached to the lower control shaft by means of a clamp, facilitating installation and servicing and also requiring a smaller hole and seal in the dash. Near the base of the steering column jacket a diecast housing encloses the lower control shaft and the inner shift levers, the latter forming shift gates for the selecting operation. A light spring at the lower end of the control shaft holds it in the down position. Within the transmission case, shifting of the gears is accomplished by two rotating forks. A cam plate is attached to each one and an interlock shaft is anchored between notches in the cams. When the overdrive is not used, Nash incorporates a longer rear housing (aluminum casting) to maintain the same position of the torque tube housing mounting. The propeller shaft and torque tube are unchanged on the Chevrolet, even though the engine is located 3 in. farther forward. This space is filled by an extension or added bearing support that is bolted to the rear of the transmission case. The front universal yoke hub is supported by a ball-indented, rolled bronze bushing pressed into the support casting. Bearing lubrication is supplied by splash from the universal joint through a drilled hole at the top of the bushing. Two holes in the web of the extension or bearing support permit lubricant from the transmission to reach the universal joint. A molded synthetic rubber seal of the limited-contact type replaced the previous leather unit to maintain the proper transmission oil level by preventing leakage of lubricant through the torque tube to the rear axle.

As for the semiautomatic transmission field, much has been accomplished since the war. The Packard electromatic clutch and the Hudson drive master can be considered temporary expedients. The GM hydra-matic,[6] introduced in 1940 in the Oldsmobile, became optional equipment on the Cadillac in 1941 and on Pontiac in 1948. The Chrysler tip-toe-matic hydraulic shift, introduced in 1942 (depending on governor action related to vehicle speed, the position of the two-range shift lever and release of the accelerator pedal to effect a shift), consists essentially of a conventional 6-gear transmission with the central mainshaft gear loose upon the shaft and an overrunning clutch member in the countershaft constant-mesh gear. Low and direct are conventional but the loose gear can drive the countershaft in lieu of the constant-mesh pinion with return to the driven shaft via the low gears to effect second speed or receive the drive from the countershaft and impart it to the driven shaft for third. The low mainshaft gear is also loose on its shaft and the hand shift actuates the clutch member, which locks this gear to the main shaft during first and second; during third and fourth the intermediate gear is clutched thereto. Oil pressure generated by a pump directly driven by the mainshaft, under the influence of the governor and relieving the driving load by torque reversal through releasing the accelerator pedal, moves the piston forward against spring pressure: the return occurs on relieving the oil. A shifting yoke attached thereto controls the movement of a clutch member splined on a hub of the intermediate loose gear, and forward movement establishes direct drive from the constant-mesh gear to the intermediate. Depending on the position of the manually controlled clutch member, the drive will be down to the countershaft for second or direct if the manually controlled clutch has coupled the internal gear to the mainshaft. A kickdown switch takes care of a down shift. The tip-toe shift was also optional on DeSoto and this year is available in the same manner on the Dodge Coronet.

[6] See SAE Quarterly Transactions, Vol. 1, October, 1947, pp. 559-565: "Automatic Transmission Control Systems," by O. K. Kelley and M. S. Rosenberger.

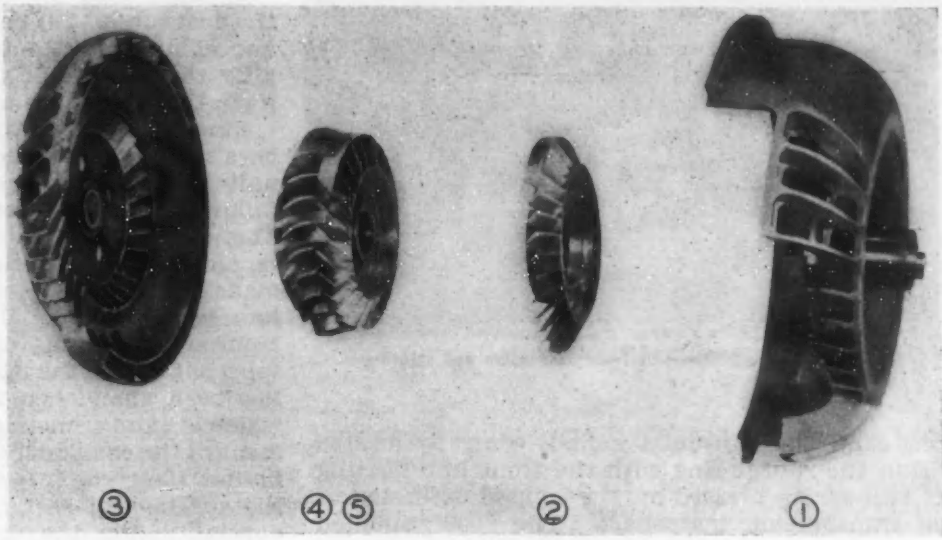


Fig. 7—Torque converter portion of Buick dynaflow drive
(1) Primary pump
(2) Secondary pump
(3) Turbine
(4) First stator
(5) Second stator

The Buick dynaflow transmission,[7] introduced on the 1948 Buick series "70" as optional equipment is now standard thereon and is currently optional on the "50." It consists essentially of a torque convertor with a stall torque of 2.25-1 and comprises a primary pump, secondary pump, turbine, and two stators, which are stationery over the low-speed range but gradually and progressively free wheel as the speed increases. (See Fig. 7.) At top speed the system functions as a fluid flywheel. Lower reduction is obtained when necessary by the low speed of a combination low-and-reverse planetary set. A direct-drive clutch, planetary band actuation, and hydraulic control are patterned somewhat after the hydra-matic. Smooth acceleration over the hydraulic phase presents a new type of passenger comfort. The production problem of the hydraulic components has not been a simple one. A new plastic mold process based on the Antioch method is used, except wax and plaster are eliminated. The basic ingredient of the mold material is gypsum or calcium sulfate. Water is added in controlled proportions and the resulting mix or slurry is pumped into the molds. The Aluminum Co. of America has developed the furnace brazing process to the point where it can make satisfactory torque convertors for heavy-duty vehicles, and it has a new casting process for the passenger-car type to provide very thin blades, which are exceedingly smooth in comparison with conventional foundry practice. It is understood that Packard will shortly introduce a torque converter and that Chevrolet is working on a steel metal version of the dynaflow.

Propeller Shaft and Rear Axle

A number of cars, such as Studebaker, Hudson, and Kaiser-Frazer, have gone to a 3-joint propeller shaft with a center bearing in order to lower the floor line without using an excessively high tunnel. With the powerplant moved forward 2¾ in., the Pontiac transmission extension has been made still longer so that the propeller shaft is shorter than before and offers no balancing problem. The same shaft is interchangeable whether the syncromesh or the hydra-matic transmission is used, since the transmissions are alike in overall length. Ford's original production vehicle and on down through the years had torque-tube drive; and it was, therefore, an event when the 1949 model, as well as the Mercury, switched to an open propeller shaft and adopted Hotchkiss drive and hypoid gears. Naturally, it has been possible to decrease the height of the drive shaft tunnel. The Lincoln axle is of the semifloating type, replacing the former three-quarter one. The rear wheels are bolted directly to the flanged axle shaft ends on all Ford-built cars, a feature originally introduced by Oldsmobile. At the same time the Nash Ambassador has dropped the Hotchkiss drive and is now using a torque tube. (See Fig. 8.) Rubber is used at the tube anchorage on the transmission as well as the junction of the truss rods with the torque tube. A tubular extension at the front end of the torque tube has a flared-out interrupted flange embedded and molded in a rubber ring, which is clamped against, and interlocks with, the transmission case extension by a pressed

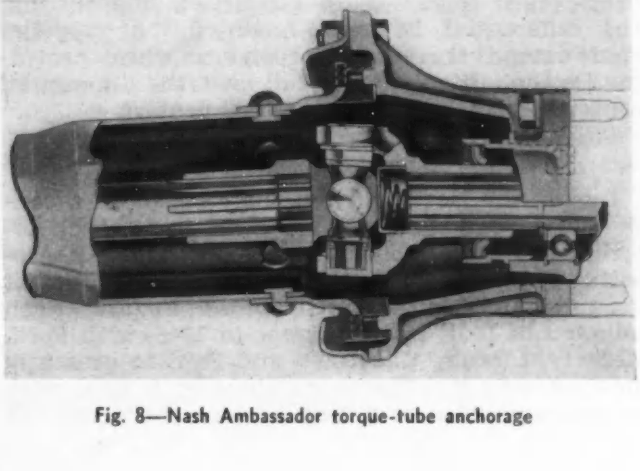


Fig. 8—Nash Ambassador torque-tube anchorage

[7] See SAE Quarterly Transactions, Vol. 2, July, 1948, pp. 477-486: "Buick Dynaflow Drive," by C. A. Chayne.

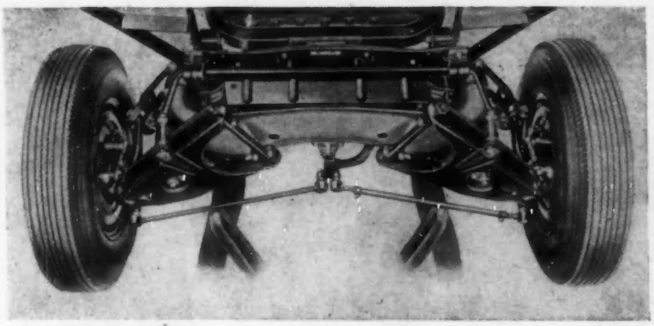


Fig. 9—Redesigned Chevrolet front suspension and steering

steel cap. The universal joint is centrally located within the rubber ring with the front hub floating on, and spring pressed by, the splined extension of the transmission mainshaft. The "600" adopted torque-tube drive in 1941 but does not have an enclosed joint. A horizontal yoke at the front end of the torque tube is supported by rubber biscuits at each side.

The Buick torque tube is slightly shorter on the series "50" and "70," due to the 3-in. decrease in the wheelbase. The strut rods have been changed, due to the relocation of the rear springs. A new torque ball boot has been designed with an improved oil seal. The same type of boot has been satisfactorily used on the 1948 dynaflow-equipped models and will be continued. The outer torque ball retainer has been redesigned as the new boot requires a retaining flange on the outer torque ball.

The rear tread, which at one time reached the high figure of 62 in., no longer needs to be excessive, since the rear seat cushion is now moved ahead of the wheel houses. Hudson provides 55½ in. (front tread 58½ in.), while Nash remains unchanged at 59 11/16 in. on the "600" and 60½ in. on the Ambassador. The Chevrolet rear tread has been decreased from 60 in. to 58¾ in. The axle-shaft wheel flanges have been reduced 23/32 in. in diameter and the number of rear wheel hub bolts reduced from six to five, since tests have shown that five are adequate. The rear spring seats are moved farther inboard on the axle housing to conform to frame changes brought about by the tread reduction.

Dodge has recently placed in production their Route-Van, a door-to-door package delivery truck, in two capacities. The rear axle is a heat-treated drop-forged steel I-beam supporting member and the differential is frame-mounted. A propeller shaft extends therefrom to each rear wheel, providing De Dion drive. Under full load, the differential center is slightly below the wheel centers.

Front Suspension

During the last few years there has been a considerable trend toward locating the front shock absorber inside the coil spring, and coaxially therewith. This originated with Monroe Auto Equipment Co. and started initially with Hudson in 1939. Nash followed in 1941, Kaiser-Frazer in 1946, with Ford, Mercury, Lincoln, Chevrolet, and Pontiac presently incorporating it. The shock absorber is in a more protected position, and Nash has currently removed the dust tubes, thus permitting cooler-running units and eliminating the tendency of the tubes to amplify road noises. Extensive tests have convinced Nash that the tubes can be safely eliminated.

Ninety per cent of the shock absorbers are of 1-in. bore and they have been made nonserviceable. The units are sealed at the factory and in the event of failure they are replaced by a complete new unit. This development came about as a result of high repair costs and inferior workmanship, which often made a poor repair job cost more than a complete new set of shocks. Units were first sealed up by Monroe in spinning over the end of the reservoir tube, but during the last year the spinning operation has been abandoned in favor of electric welding, which is more economical and provides better alignment of the components. Powdered-metal parts are finding their way into shock absorbers as a result of the shortage of cast iron. Due to their excellent durability and the saving made possible by the elimination of machining operations, powdered metal has completely replaced cast iron in both the piston and the rod guide.

Chevrolet has redesigned its front suspension by using stampings instead of forgings in the upper and lower wishbones, effecting a 12% reduction in weight. (See Fig. 9.) The axes of the wishbones now converge to the rear and meet a short distance behind the front cross-member in place of their previous parallel disposition. The chrome-alloy steel springs are shot-peened and made from heavier stock with a deflection rate of 340 lb per in. as against 300 last year, due to their increased front loading. The springs are enclosed in a welded pressed steel housing or tower, ⅛ in. thick, each closed at the top and welded to the ends of the suspension cross-member ends. The upper spring seat is welded to ledges formed in the tower. The complete upper wishbone is a one-piece steel stamping ⅛ in. thick. Its pivot shaft is force-threaded into reinforced holes in the spring housing tower. Threaded bushings are also force-threaded into extruded holes in the new arm where it is pivoted on the stationary shaft. The lower wishbone is also a single stamping with the central area formed to provide the lower seat for the coil spring. Pontiac continues with the stereotyped wishbones, the upper one being anchored to a forged pivot shaft bolted to a low tower riveted at each side to the top of the frame cross-member. An inverted bell-shaped cap positions the shock absorber top threaded stud. Rubber grommets insulate the shock absorber at this point and at the lower wishbone anchorage, also permitting angulation. Chevrolet uses a similar grommet mounting.

The lower wishbone fittings on the Chrysler-built cars are easier to reach for lubrication because of a new offset in the sway bar. To provide additional lubrication the upper control arm is equipped with a lubricant reservoir. Whereas Nash welds together the top channel section wishbone members back to back to form an H, the Ford-built cars, and most others, have the flanges welded together to form a box section. The Nash "600" spring has a natural oscillation rate of 70 cycles per min, the Ambassador 74; both with a deflection rate of 108 lb per in.

Another Monroe development is the use of the

linkless design sway bars. In the Ford, Chrysler-built cars, and Kaiser-Frazer, the end of the bar is anchored in a rubber block clipped within the forward channel section of the lower wishbone. In the Mercury and Lincoln, the sway bar end terminates in a rubber bushing located in a capped boss on the front wheel spindle support forging.

The front tread on both Nash models is 54 11/16 in. as against 57½ in 1948. The turning radius is better than in 1948, being 20 ft 2 in. on the "600" and 21 ft 4 in. on the Ambassador. Chevrolet's front tread is 57 in., ⅝ in. narrower than last year.

The axles of the Ford-built cars now have hypoid gears. Spicer is supplying the Mercury and Lincoln axles. All Spicer axles have a scraper that collects oil from the top of the revolving ring gears and flows it through a passage terminating between the two pinion bearings. A constant level is maintained by the height of a return flow channel. There is also a return channel to take care of any excess oil that might work through the front bearing and menace the pinion shaft oil seal.

Rear Suspension

The location of the Buick rear springs on the series "50" and "70" has been changed from in front of the rear axle to over the axle housing but slightly offset to the rear. This change reduces the reaction on the rear engine mountings and permits more freedom in their design to obtain the correct frequency. Rear springs have been changed with respect to spring rate and load to provide correct weight distribution because of the change in spring location and wheelbase, and to ensure an improved ride and an easier handling car. Nash rests the bottom seat of the rear coil spring centrally over the top of the axle housing. Nash engineers claim no weight sacrifice accompanies their design of suspension and drive. Coil springs are much lighter than leaf (approximately 10 lb each, compared to 40) and practically all of this saving is absorbed by the torque tube and the track bar. The oscillation rate of the Nash "600' spring is around 82 cycles per min, the Ambassador 84. Both have a deflection rate of 125 lb per in. Chevrolet runs 108 lb, with its 7-leaf spring, as against last year's 115-lb 8-leaf one, coupled with a weight saving of 9% and a softer ride. Mercury and Lincoln have liners between the rear spring leaves and rubber bushed shackles and spring eyes, and with no track bar.

Most cars have adopted what is known as the "sea leg" mounting for the rear shock absorbers. The top mounting pin is closer to the center of the car than the lower one so that the shock leans in at approximately a 30-deg angle. This has been found beneficial in providing lateral stability for the rear end and in some instances made possible the elimination of a track bar. Pontiac was the first to use this type of mounting and others now using it are Hudson, Kaiser-Frazer, Chevrolet, Packard, the Ford-built, and the Chrysler-built cars. Chevrolet shock absorbers are attached at the top to a heavy gage steel reinforcement welded to the body floor just back of the rear seat at the frame kickup. The shackled ends of the rear springs and the forward spring eye are rubber bushed instead of using threaded bushings and the shackle angle has been increased to 95 deg.

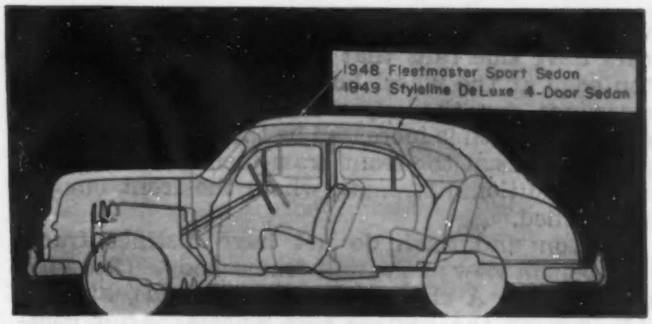


Fig. 10—Forward movement of engine and seating, and lower silhouette, as exemplified by Chevrolet

On the Chrysler-built cars an extra rubber bumper is provided above each rear spring to the rear of the forward spring eye. It is of extruded rubber, attached to the frame by means of a clamp, and limits the movement of the springs to assure a smoother ride, especially during fast acceleration. New U-bolt plates are attached to the bottoms of the rear springs with chamfered ends.

Torsion-bar springing, front or rear, has not yet reached the passenger-car field. The Goodrich Torsilastic spring was incorporated in an experimental car[8] shown at the 1946 SAE Summer Meeting, bristling with installation novelties.

Frame

Most front cross-members are now V shaped in view of the forward location of the engine (Fig. 10) and so as to provide clearance at the center to permit dropping the sump pan. In the early postwar cars it was sometimes necessary to remove the engine from the chassis to perform this operation. The Chevrolet front cross-member of semitubular construction is entirely new, incorporating the suspension spring towers at each end. It sweeps back to the rear at an angle of 27 deg 40 min at each side, the semicircular upper portion being 0.140 in. thick and the bottom plate 0.090 in. As in previous years the cross-member is bolted to facilitate removal when necessary. Flanged saddles are riveted and welded to the cross-member and 12 bolts are used at each side to secure the cross-member to the side rails. They continue in the form of a box girder construction with the channel portion and bottom plate being 3/32 in. thick. Formerly, the bottom plate was ⅛ in.

The Buick X-member has been lowered ¾ in. on all frames to make room for the new rear seat foot rests. Convertibles and estate wagon models have X-members made of heavier stock for added strength not imparted by the body. Because the engine and radiator units are moved farther forward, an extra radiator support cross-member has been added to the Chevrolet and Pontiac frames,

[8] See "Independent Four-Wheel Suspension Using Rubber Torsion Springs," by A. S. Krotz, R. C. Austin, and L. C. Lindblom. Digested in **SAE Journal**, Vol. 54, November, 1946, pp. 34-41 under title "Car Design Emerges Using Torsion Spring."

which also braces the front end. The Chrysler-built cars have side rails that are chamfered along the top edge at the wheel housings to provide greater clearance for the super-cushion tires on turns. Frame strength is increased by the use of additional gusset plates at the front cross-member, which also afford additional strength where the front bumper is attached.

Hudson and Nash do not have distinct frame members in view of the unit frame-body. (See Figs. 11 and 12.) A channel member extends, at each side, outside the Hudson rear wheels. After completion of the body structure, the front channels, on which the powerplant and suspension have already been assembled, are joined to the frame-body main sills by four cold-sqeezed rivets at each side, followed by arc welding.

Controls

While there do not appear to be any basic changes in steering gears, improvements have been brought about by careful engineering in the application of this unit. Balances in the joint use of wheel suspension, shock absorbers, sway eliminators, tires, linkage, and steering gears are providing better steering today. In a number of cases, steering wheel rim diameters have been increased. Steering wheels are either of the 2- or 3-spoke design, with the upper half clear for instrument vision. One-half or three-quarter horn rings are most popular, although occasionally a full circle is retained, as in the case of Pontiac and Chrysler-built cars. The Nash and Pontiac have slightly less than 180 deg between the two spokes, measured toward the operator.

Nash provides a massive steering column jacket in the form of a 4-in. tube, which conceals the steering shaft, gear shift rods, speedometer cable, the wiring leading to the Uniscope instruments, the light switches, and the directional signal. All Buick series "70" models have a recirculating ball-type steering gear with 23.6-1 ratio for improved handling, replacing the former 19.8-1 gear. A new steering column support has been designed for the series "50" and "70" models; shims have been incorporated to allow for up-and-down adjustment of the column. A new brace for the column support has been designed as a part of the body structure. Pontiac's column has been moved out 2 in. farther from the center of the car. Chevrolet's wheel is located at a lower, more natural angle, improving visibility as well. Two diagonal braces in the form of 5/16 in. rods extend from the lower flange of the instrument panel to the dash and replace the former single brace. The worm shaft bearing is now adjusted by a nut at the base of the steering housing, which contains the lower tapered roller worm bearing and the correct transverse adjustment of the straddle-mounted pitman shaft is taken care of by the coaxial, enmeshed circular head of an adjusting screw. A short drag link runs forward from the steering gear pitman arm to an idler lever supported on the front cross-member by a forged bracket, whose axis is mounted at a slight angle from the vertical to equalize more nearly the loading on its upper and lower bushings. The idler lever pin oscillates therein. The axes of the front suspension lower wishbones intersect at a point closely adjacent to the inner ball joint centers of the tie rods, as in the case of the original center point steering controls that accompanied individual wheel suspension. The right tie rod is slightly longer than the left.

The Cadillac is easier to steer since the new powerplant is 200 lb lighter. To minimize shocks, the Ford-built cars have the pivot points of the tie rods located on the same centerline as the wheels. The steering worm is toward the center of the car, whereby the pitman arm swings in a plane parallel to the steering column axis (as viewed in elevation). A steering idler arm is symmetrically located on the right side rail and a link connects the two arms. A tie-rod takes off to each wheel spindle arm from the connecting link at approximately 30% of its length from each end. The Studebaker Champion wheel has been increased to 18-in. diameter. As in the past, the position of the steering wheel can be altered to suit the size of the driver by changing the mounting bracket.

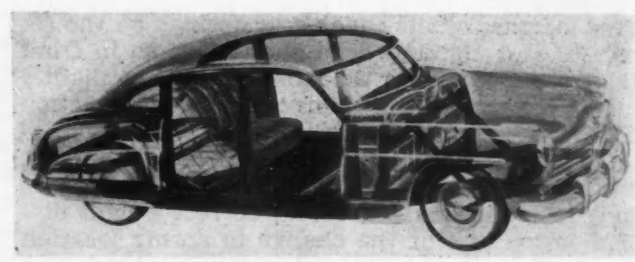


Fig. 11—Hudson monobuilt body and frame

The Wagner self-centering brake relies on the rounded shoe ribs rolling upon the anchor block, whose sides extend radially to the axle center. In the Studebaker application, self-adjustment is obtained by a contact plug located near the center and projecting through the forward shoe. Upon brake application, the plug is held against the drum by a spring that resists its inward movement. As the lining wears, the plug is depressed farther into the shoe assembly and this movement forces the adjuster lever to pivot away from the shoe table and adjuster wedge. Clearance thus formed between lever and wedge is then automatically taken up by wedge-actuating spring forcing the wedge to move upward. The Al-Fin Division of Fairchild Engine & Airplane Corp. has produced bimetallic brake drums with aluminum-bonded fins to cast-iron liners (same process as Continental's recent license for bonding fins to aircooled cylinders), producing a noticeable reduction in fading due to lowered temperatures, reduced weight, longer drum and lining life with no checking of the drum.

Since bonded brake linings proved themselves on the lighter Dodge and Chevrolet trucks, the Chrysler-built line is now so equipped, as well as Chevrolet. In the latter case the front brakes now exert 57.7% of the total braking effort as against the previous 52.5%, brought about by the new seating arrangement and engine location. This has been ac-

accomplished by changing the front wheel cylinders from 1¼ in. to 1 5/16 and the rear decreased from 1 3/16 in. to 1⅛. Crosley has also joined the ranks of bonded linings. A new 3-part brake wheel cylinder piston is used in the Chrysler-built line to provide more piston seal and prevent leakage of brake fluid. It consists of piston, expander, and rubber cup. Pressure from the master cylinder forces the spring against the expander, pressing it into the rubber cup, and expanding it. A light spring is used in front wheel cylinders and a heavier one in the rear.

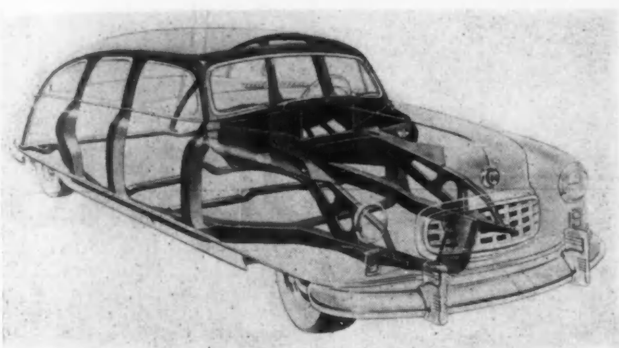


Fig. 12—Nash unit frame and body

Weight shift has caused Lincoln to increase its front brake shoe width from 2 in. to 2¼, the diameter remaining 12 in., giving an effective total service brake lining area of 220 sq in. Pontiac's 11-in. brakes have 2¼-in. front and 1¾-in. rear linings, as against last year's $2 \times 1\frac{3}{4}$ in. ones. Last year's Oldsmobile Futuramic initiated $2\frac{1}{2} \times 2$ in., which it still continues. The other models were $2 \times 1\frac{3}{4}$ in., which remain in the series "76" but change to $2\frac{1}{2} \times 2$ in. in the "88". Hudson has $2 \times 1\frac{3}{4}$ in. The industry is beginning to realize the desirability of obtaining maximum retardation from each wheel. However, they must lock at no time, should the road coefficient of friction drop, such as is presented by a surface covered with rain, snow, or ice. To prevent such a contingency, Wagner Electric Corp. has brought out a ratio changer whereby, under manual control, the rear braking effort remains unchanged but the front is reduced to one-half. This is done hydraulically by introducing a stepped piston with 2-1 areas in the front brake line.

The Buick step-on parking brake has 2-in. increased clearance between its pad and that of the clutch pedal; the clearance between the cowl kick-pad and the parking brake pad has been increased 1 in., allowing more space for operation and decreasing the possibility of interfereence. The foot pad swings outward in a plane making a 6-deg angle with the centerline of the car. Release is now obtained by pulling a knob located on the lower left corner of the instrument panel.

There has been a sudden rejuvenation of the cane handle parking brake control that was originally found on the 1934 DeSotos and Chryslers. In adopting it and in view of the rigging leverage, Chevrolet has gained a mechanical advantage of approximately 22% over its previous system. It is located to the right of the steering column and a 60-deg turn of the handle releases the brake, which is retained by two pawls not spaced an even mutliple of the pitch of the ratchet teeth on the control rod. Studebaker provides a pushbutton release built into the handle axis to prevent accidental release. Pontiac's system is similar to Chevrolet except a T-handle is used. The Chrysler-built cars retain the depending lever but provide a double pawl mechanism, reminiscent of the Hudson application, to the former central brake lever. The parking brake cables of Pontiac and the Ford-built cars are protected by a rubber seal, where they enter the conduits ahead of the rear axle to prevent entrance of water and foreign matter.

The Chrysler-built cars have an accelerator pedal hinged at the heel and attached at this point to the top of the toe board. It eliminates the friction of the customary throttle rod, where it projects through the floor seal.

Wheels and Tires

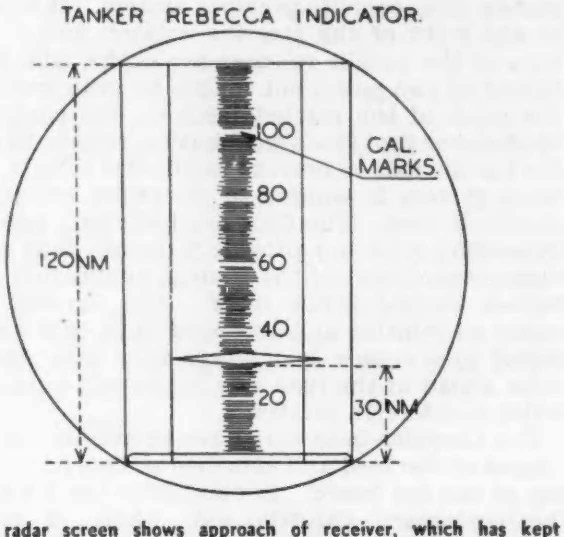
All cars are using 15-in. wheels with the exception of Crosley, which uses 12-in. ones. The outstanding change has been the trend to lower pressure tires.[9] It was virtually an oversizing program on a 1-in. smaller rim diameter to operate at 4-psi lower pressure than the regular balloon tires. The industry has adopted the program as shown in the Tire and Rim Association Handbook in which the ratio of rim width to tire section is approximately 70%. Supplementing this setup as permissible practice in each size is a rim ½ in. wider (approximately 77% width-section ratio).

With the high powering of most cars and lighter loading on the rear tires, the problem of good traction has arisen on wet pavements and particularly on ice. One attempt involves curing a high carbon steel coil spring in the ribs of the tire tread. As the tire and spring wear, the interrupted convolutions of the spring present sharp traction points, as in the case of the Midgely tread of around 1910. Tires are sometimes recapped with camelback in which saw dust, salt, nut shell fragments or other material is mixed into the stock, with the idea that as the tread wears these particles will fall out and make small cavities which create a gripping action. The Goodrich tubeless tire is now in production. It has an inner layer of butyl rubber to hold the air as would an inner tube. A series of six small concentric ridges is located around the bead to seal it at the rim contact. Puncture sealant covers the inner circumference. Butyl rubber inner tubes have unexcelled air retention, but their manufacture was suspended, due to their hardening and cracking, during the extraordinary winter weather of 1947-1948. They are, however, beginning to come off the production lines again in view of the revised cold processing of rubber.

(Part II, which covers the remainder of this paper, will be published in the June issue.)

[9] See "Application of Extra Low Pressure Tires for Passenger Cars," by W. E. Shively. Digested in **SAE Journal**, Vol. 56, March, 1948, pp. 22-23, 28 under title, "Announce Facts about New Softer Tires—What They Are."

TANKER REBECCA INDICATOR.



Tanker's radar screen shows approach of receiver, which has kept tanker base informed of estimated time of arrival and fuel required. Tanker is homing on receiver's "Eureka" radar beacon by means of "Rebecca" transponder beacon

BASED ON PAPER* BY

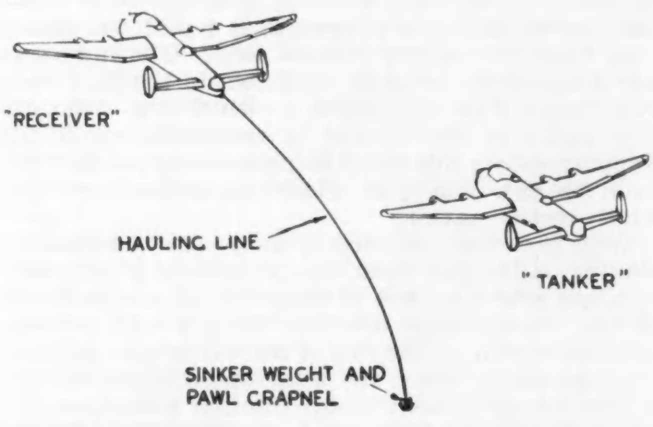
C. H. Latimer-Needham

Chief Engineer,
Flight Refuelling Ltd., England

* Paper "Refuelling in Flight" was presented at SAE Annual Meeting, Detroit, Jan. 13, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

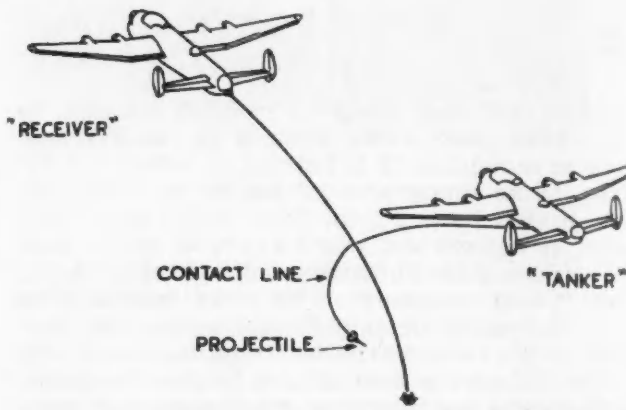
ADDING

1



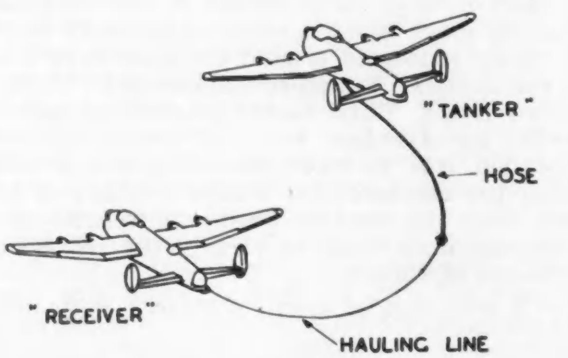
Tanker intercepts receiver at rendezvous. Receiver trails hauling line with sinker weight at end. Tanker comes into position on rear starboard quarter, a little below receiver

2



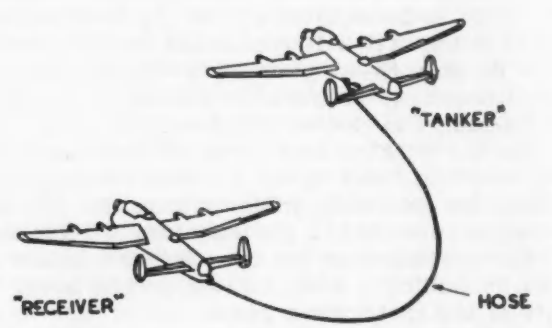
Contact line passes forward of hauling-line arc. Tanker fires contact line from line-throwing gun when tanker reaches level of hauling line suspended from receiver

5



Receiver operator winds back hauling line bringing hose. Hose nozzle approaches reception coupling in tail of receiver. Tanker pilot climbs to about 70 ft above receiver, if system relies on gravity flow

6



With hose nozzle locked in position, tanker operator flushes system with nitrogen to replace air with inert gas. Then main fuel cock is opened, and fuel flows at rate of about 120 U. S. gal per min. On completion of fuel transfer, system is flushed again

FLIGHT refuelling of commercial aircraft on long-distance operations offers savings in capital and running expenses that far outweigh its cost.

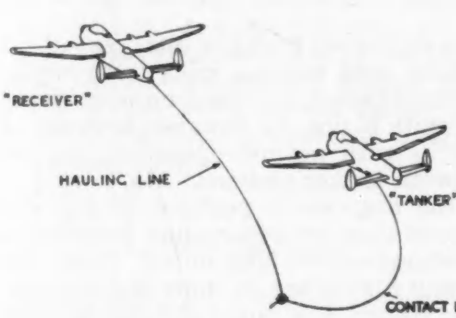
Almost any airline route in the world can be operated with flight-refuelled aircraft of 3000-mile range—the design range which happens to correspond to the most efficient division of disposable load into fuel and payload for the newest transports.

Because of the short range requirement, flight-refuelled aircraft can be smaller and therefore less costly than those designed to cover a long route without flight-refuelling. Fuel loads are lighter because the airplane and its range are smaller. And the ability to refuel in flight cuts down on the fuel reserves needed as protection against unfavorable winds.

Flight-refuelled aircraft will cost less to operate, as well as less per pound to buy. Elimination of

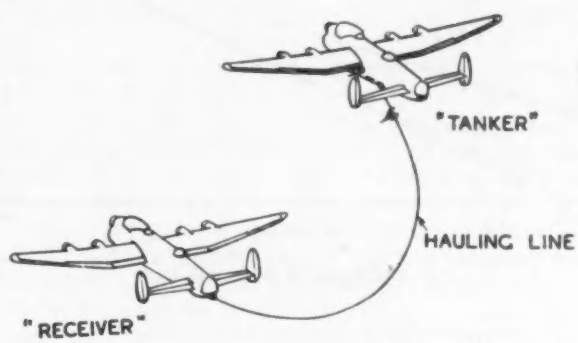
FUEL IN FLIGHT

3



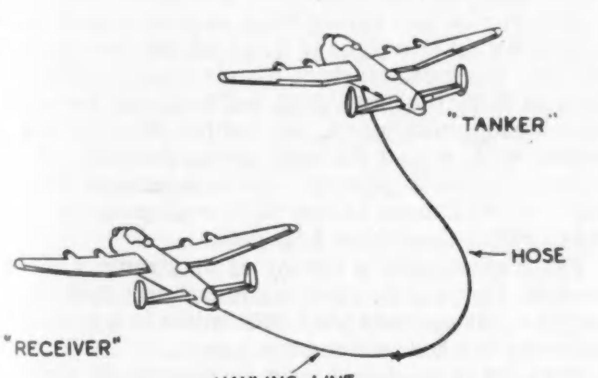
Contactor at end of contact line is retained by pawls of grapnel at end of hauling line after contact line has made contact with and slid down hauling line

4



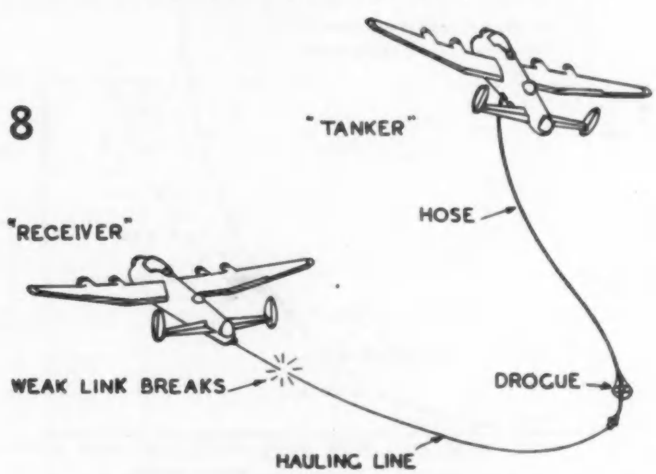
Tanker operator winds in contact line by hydraulically operated winch, removes sinker weight and grapnel from hauling line, and attaches hauling line by bayonet coupler to hose nozzle

7



Receiver pays out hose and complete hauling line. At end of hauling line is short "weak link" connecting hauling line and weak-link line. Weak link and about 50 ft of weak-link line also paid out

8



Weak link snaps when tanker veers to starboard. Tanker operator winds in hose and attached hauling line. Drogue steadies hose. Receiver operator winds in weak-link line. Process takes about 30 min for transfer of 2880 gal

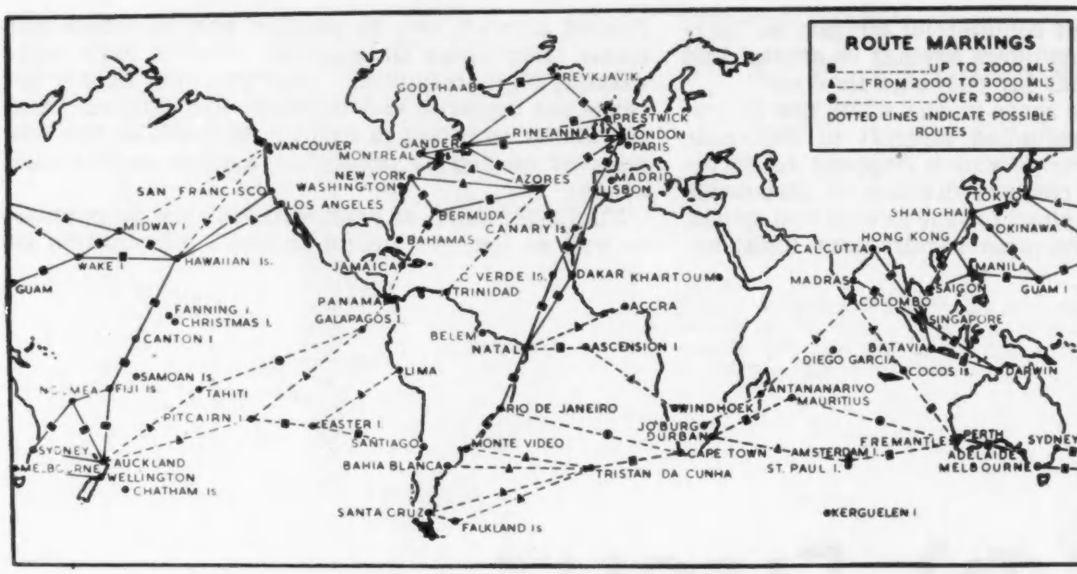


Fig. 1—World's over-water routes

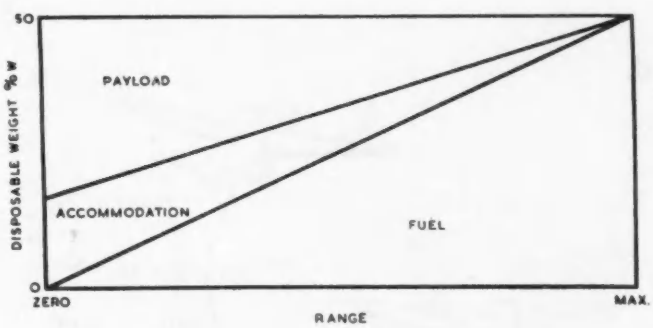


Fig. 2—Division of disposable load

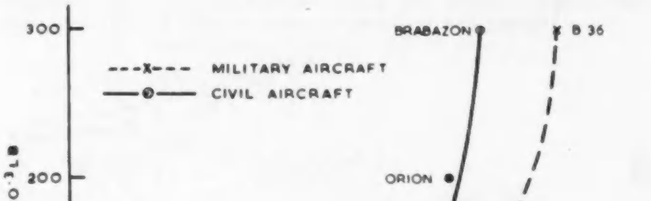


Fig. 3—Ultimate theoretical range of aircraft

landings and take-offs made solely for refuelling will reduce time on the route, wear and tear on the engine, and chances for accident. Landings made for passengers can be made on shorter, cheaper runways.

Yet it is estimated that the total cost of a refuelling in flight need not run more than about $1000.

The critical factor in a long nonstop flight-refuelled air route is not the distance between the two terminals as with normal operation, but the distance between tanker stations. Fig. 1, on which are mapped the over-water portions of the main air routes, shows that few separations between possible tanker stations exceed 3000 miles. Since flight refuelling may take place at some distance out from the tanker station, the range of flight-refuelled airliners need not be more than two-thirds of the separation—or allowing for unfavorable winds and other operating conditions, 3000 still air miles.

This 3000-mile figure for range turns out to be an almost magic number; it is the ideal range from the standpoint of distribution of a modern aircraft's disposable load between fuel and other load.

Experience has shown that, assuming 50% of the airplane's gross weight to be available for disposable weight, the most efficient distribution of disposable load is 50% fuel and 50% payload (or passengers plus accommodations). Of course, the distribution varies with range for any given design. At zero range, all the disposable load is available for payload; at maximum range, all the disposable load is fuel. Fig. 2 illustrates this point.

Fig. 3 represents a survey of maximum ranges of modern transports, carrying maximum fuel and no payload. It appears that 6000 miles is a reasonable achievement for passenger aircraft.

If Fig. 2 is relabeled to show maximum range as 6000 miles, as has been done in Fig. 4, the optimum distribution of disposable load falls at 3000-miles range—the magic number again.

Fig. 4 emphasizes the rapid diminution of useful load with increase in range. The aircraft whose maximum range was 6000 miles could devote 16 2/3% of its gross weight to passengers at the 3000-mile range but less than 6% at 5000 miles.

If payload is kept constant, gross weight obviously will increase with range. Fig, 5 shows how fast gross weight increases. This plot has been developed from Fig. 4, on the assumption of constant payload. It shows that an airplane for a 3000-mile flight need be only one-third the size of the airplane for a 5000-mile flight.

By restricting range to 3000 miles, flight refuelling obviates the need for giant aircraft to carry the fuel. Aircraft for long flights need not be flying fuel tanks. And smaller fuel reserves will suffice. Unexpected strong headwinds can be compensated for by increased fuel transfer.

Flight refuelling can circumvent another source of inefficiency now experienced on transatlantic runs: the prevailing westerly winds. Without refuelling, an airliner capable of carrying a full complement of passengers on the easterly trip with the usual following winds may have to sacrifice payload to fuel reserve on the return trip. But careful spacing of refuelling points and amount transferred can equalize capacity. For example, the eastward flight from New York to London could be refuelled over Gander and the westward flight over both Shannon and Gander.

All this saving in range and weight leads to aircraft that cost less per pound to buy. Since 3000-mile-range flight-refuelled airplanes could be used on any long route, standardized designs for 50, 75, and 100 passengers would spread design, development, and some production costs over a large number of airplanes produced.

Landings Eliminated

Once purchased, flight-refuelled aircraft will continue to effect savings. In a field where the value of speed is high, flight refuelling eliminates the delay of landings for the sole purpose of taking on fuel. Eliminating two 1-hour stops from a 20-hr journey is equivalent to an 11% increase in speed. Besides, elimination of landings saves wear on the engine during take-off and on undercarriages, wheels, tires, brakes, and flap gear during both take-off and landing. In experiments, flight-refuelled BSAA and BOAC airplanes have required remarkably little maintenance.

By eliminating landings and take-offs, where roughly 70% of accidents take place, flight refuelling improves safety. In case an accident does occur, the small quantity of fuel aboard reduces the fire hazard.

Adoption of flight-refuelling could stem the trend toward longer, heavier, more expensive-to-build-and-maintain airport runways, because flight-refuelled aircraft are lighter for a given payload. Such aircraft could even take off from short runways with light fuel loads, then receive the bulk of the fuel load in flight at operating altitude.

Although runways would be required for the tanker aircraft which accomplish the refuelling, there would be no need at tanker stations for the passenger accommodations needed if refuelling takes place on the ground.

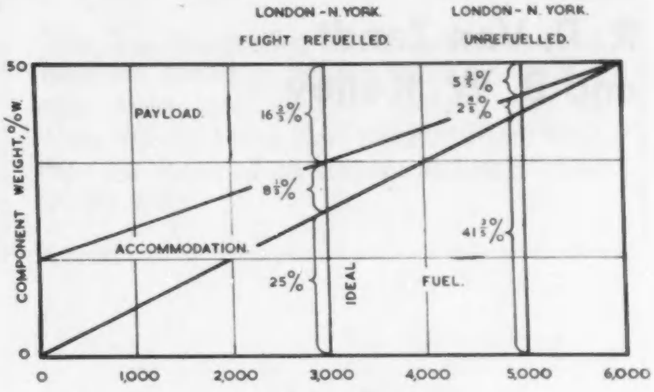


Fig. 4—Relationship of payload to range

Refuelling Cost Moderate

In exchange for all these economies in capital and running expenses, there is the cost of the flight-refuelling operation, which depends on the frequency of operation and the tanker's duration of flight.

Indications are that if frequency of operation is at least two flights per day, refuelling with a 65,000-lb tanker will cost about $1000 per refuelling. This allows for a radius of operation of about 100 miles. Increments of radius would increase cost by about $120 per 100 miles.

In the case of one operation per day, if two aircraft were maintained at the tanker station, costs would total about $1600 or $1800 per operation.

Since the average airline landing and take-off together cost an estimated $7.30 per 1000 lb gross weight, a refuelling in flight at $1000 costs about the same as a landing and take-off for a 135,000-lb airplane. All other benefits of flight refuelling are gratis.

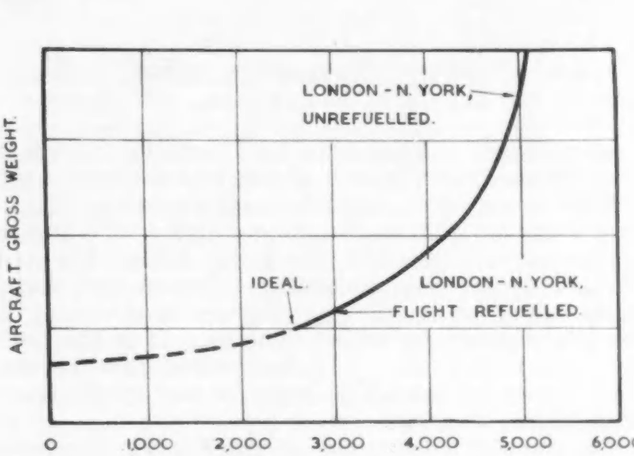


Fig. 5—Effect of range on aircraft size with constant payload

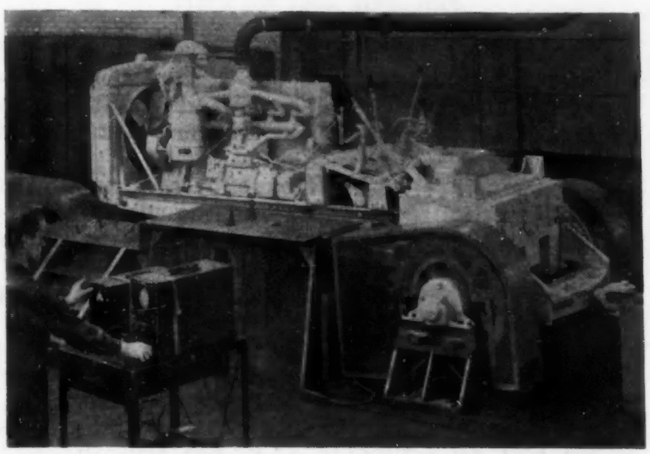
BASED ON PAPER* BY

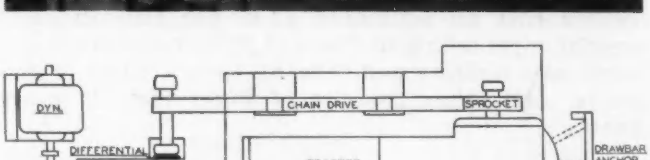
R. P. Van Zandt
and B. W. Kelley

CATERPILLAR TRACTOR CO.

TESTING TRACTOR

1. Chassis Dynamometer





Our closest approach to field tests in the lab is the chassis dynamometer, at left, which involves the whole tractor, except for the track assembly.

We use the chassis dynamometer for such things as an overall check of the power train of a new design, or for any particular parts of the drive. Although used mostly for crawler tractors, it is flexible enough for wheel vehicles. It is the only machine in the laboratory that can endurance-test bevel gears or final drive gears of any model traction vehicle we make.

On the test tractor, the regular tractor sprockets are replaced by special ones of the same pitch diameter to accommodate a pair of smaller pitch, heavy-duty chain which are easier to work with than the heavy, clumsy track chain. A spur gear differential on the jack shaft divides the load equally between these two chains; power flows from here through a higher speed chain to a speed increaser, and finally to an eddy current dynamometer.

Tractor weight is supported as in actual operation, with the tracks in place. These supports permit free fore and aft motion and offer a minimum of restraint to the pull of the chains, which is taken up by the anchored drawbar at the rear to simulate drawbar operation of the tractor. Load through the tractor gearing is measured either by pull of the drawbar or by the beam scale reading of the dynamometer.

Corrections must be made for the pull on the slack side of the chain in the former case and for loss in the chains, jack shaft, and speed increaser in the latter. If desired, the drawbar can be disconnected and the chain pull resisted at the sprocket shafts to simulate dozer operation (under steady load). Safety shutoff devices are used to stop the engine if a drive chain breaks, or to disengage the flywheel clutch if the restraining means should fail and otherwise permit the tractor to move forward on the chains.

In final drive gear tests, it is possible to increase the load out one side by using a special set of pinions on the spur gear differential, which will unequally divide the load on the chains to a 70-30% proportion, or by locking the differential and taking all the driving load out through one chain. When this is done, the drawbar is swung over to one side to balance as far as possible the unequal pull on the

* Paper "Gear Testing Methods for the Development of Heavy-Duty Gearing," was presented at SAE Annual Meeting, Detroit, Jan. 11, 1949. It has been printed in full in SAE Quarterly Transactions, April, 1949. (This paper is also available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

GEARS

The four lab testing techniques described here are aimed at helping build up tractor gear know-how, so that designs may be closer-to-right the first time and less time may be required to correct failures found in the field.

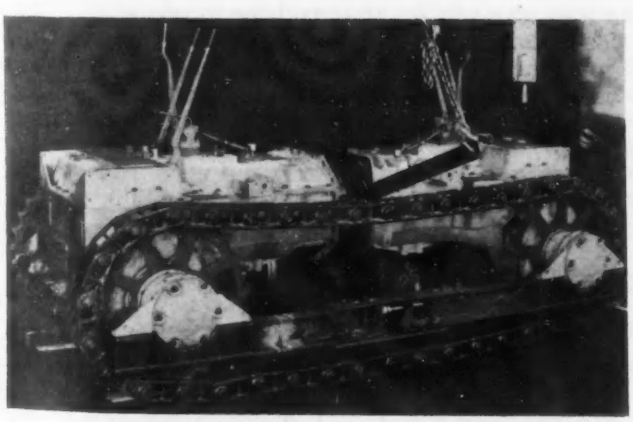
sprockets; otherwise this would tend to make the tractor yaw and throw excessive side loads on the support.

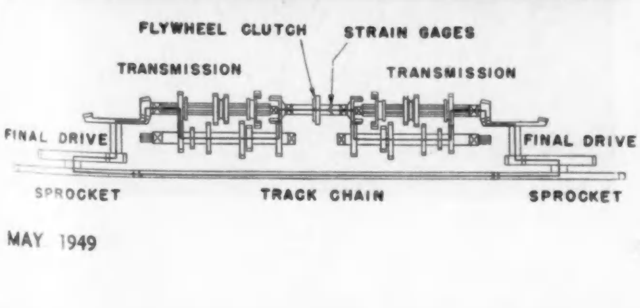
Main limitations of the chassis dynamometer are: (1) high cost and space requirements; (2) waste of fuel and power; (3) inability to overload the train (except final drive) and to accelerate the test without danger of the engine or some other component not under test breaking down; and (4) expense of tear down and assembly of new test parts.

2. Opposed Tractors

Below is shown another test rig which involves the whole tractor, but requires much less driving power and a minimum of special equipment. Engine and dynamometer are eliminated.

Two tractor main frames with gearing are tied down on a base and opposed in such a way that they can be linked together by a common track chain, connecting the two opposed sprockets at one end, and by a flywheel clutch that units the two transmission driving shafts at the other end. This encloses the power circuit within the two machines so that both are pulling against a common clutch and chain.





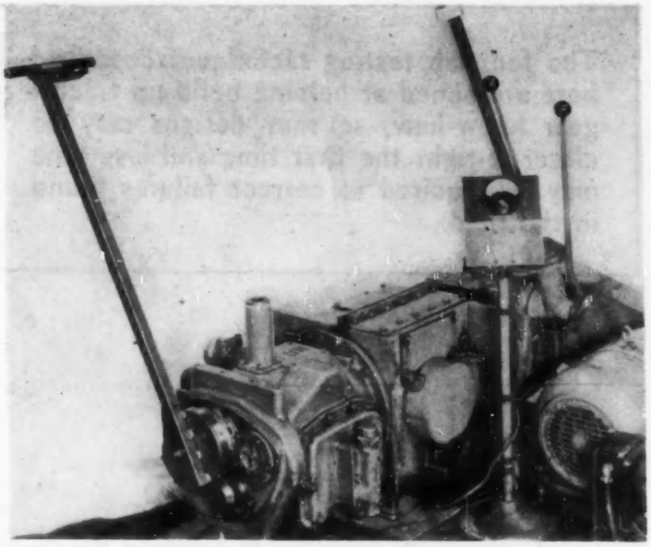
Both units are loaded by disengaging the flywheel clutch, putting a twist on the two driving shafts with suitable wrenches to develop required torque, and by locking this torque into the system by engaging the clutch. This machine is driven by V-belts from a 20-hp electric motor, connected to the transmission driving shafts. Only enough power to overcome the machine's friction loss is required.

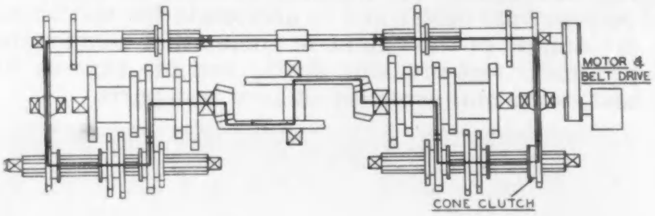
Electrical strain gages on one of the driving shafts measures torque in setting this machine. Any change in torque due to wear in parts—such as gears, bearings, and chain—after a certain amount of running, can be determined by releasing the load and noting the difference in strain gage output. To continue the test, the load must be reset.

Because the machine was made up for a final drive gear test, only the final drive on one side of each unit is loaded. As in the chassis dynamometer, this makes it possible to overload the final drive without overloading the rest of the drive and risking failure of some other part. To load both final drives of each tractor would require another chain and torque-dividing means, complicating the setup.

Direction of loads in gears and bearings is the same as in a regular tractor, but direction of rotation is reversed in one unit. Thus, tests simulating a tractor drive are valid only on one unit; the other is operating as though the tractor were going backwards and the engine acting as a drag or brake. For this reason we do not expect to compare final drives of the two units, but will limit testing to one and consider the other as a loading means.

3. Opposed Transmissions





Shown at left is an example of the same closed cycle principle applied to a pair of transmissions running back-to-back.

They are joined at the output end by a special cross shaft carrying bevel gears, interconnecting the two bevel pinion shafts, and at the input end by a couple joining the two top shafts. Load is applied as with the opposed tractors—by a pair of wrenches on one of the transmission shafts. The power train is uncoupled by a special gear, which is joined at its hub through a tapered connection. This in effect becomes a cone clutch which is pulled up to lock the load by means of suitable bolts.

Again only enough power to overcome friction is required to drive the two transmissions.

Although it consists mostly of production parts and is cheap to build, it has serious limitations. First, the load is hard to apply and hold. Second, no particular part of the train can be overloaded without overloading the whole unit. However, we have found it very useful for running oil stability tests and wear tests, evaluating the effect of dirt in the oil on gear and bearing wear.

4. Single-Pair Gear Test Machine

None of the first three machines is suited for general gear fatigue tests. A machine developed for this work, shown below, is called a single-pair gear test machine because it tests a single pair of gears at a time.

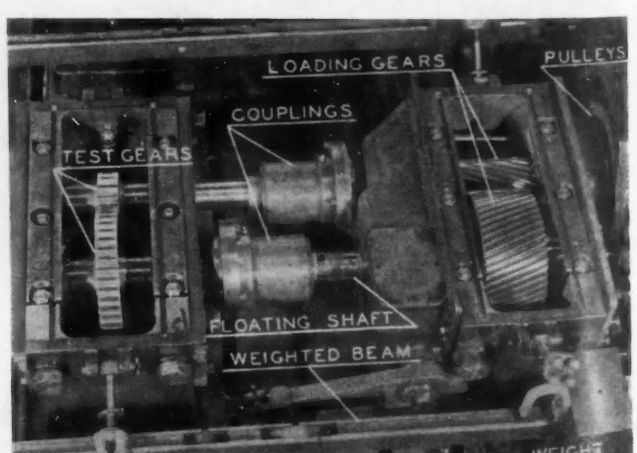
Load and speed readily can be varied and controlled; test gears easily can be changed; it is sturdily constructed so that only the test gears will fail (with occasional embarrassing exceptions); test gears can be inspected easily; and it is protected with safety shutoffs so that it can be run with minimum attention.

This design also uses the closed power circuit so that only friction loss must be overcome to drive the machine.

Basically, it is two gear boxes connected together by couplings—one box housing wide-face nitrided precision loading gears and the other the test gears. Loading gears are helical. One is mounted on a shaft that can float axially, the other is on a fixed shaft. Both test gears are mounted on axially fixed shafts.

Easiest way to understand how load is applied is to imagine the machine stationary. Any axial movement of the floating shaft will cause that loading gear to twist or wind up on its mate due to helix, which in turn is resisted by the test gears after all the backlash is taken up. Any thrust load on the floating member imparts a corresponding tangential load on the gear teeth, balanced by an equal tangential load on the test gears.

Now imagine the gears in the machine being driven at some convenient point by a motor and pulley and you see the machine in operation. Axial load is applied to the floating shaft through a thrust yoke and a system of levers, which end up in a beam carrying a sliding weight. Position of this weight on the beam establishes the load on the machine.



PROGRESS

in Personal Aircraft

BASED ON PAPER* BY

Herb Rawdon

Assistant Chief Engineer
BEECH AIRCRAFT CORP.

PERSONAL aircraft currently in production feature aerodynamic cleanliness outside and styling and comfort inside.

Postwar designs have had the benefit of extensive wind tunnel testing to improve aerodynamic form. And small controllable-pitch propellers are available, which make small high-performance airplanes practical.

Although the predicted conversion to all-metal construction did not materialize in all aircraft produced after the war, some all-metal personal aircraft did appear. And metal replaced wood and fabric-covered structures in other aircraft. There is a tendency toward all-metal construction and simplified form for control surfaces. Luscombe and Aeronca use all-metal wings with single-brace struts. Other manufacturers rely on fabric-covered wood- or steel-framework wings but build all-metal fuselages, either monocoque structures or combinations of longitudinal stringers and bulkhead frames.

Businessmen are the target of postwar designs. Interiors reflect the stylist's touch, and seats are built to be comfortable.

Manufacturers who did not feel justified in producing completely new postwar designs cleaned up their prewar designs aerodynamically, thereby improving performance and efficiency. They restyled interiors, improving comfort and convenience as much as dimensions of the old designs permitted. Cost prohibited redesigning for all-metal construction, but the conventional fabric-covered welded-steel-tube construction offers ease of maintenance and repair, in compensation.

Propeller blades for personal aircraft are almost universally wood with an overlaid metal tipping. The metal tipping yields appreciably better performance where tip speed exceeds 850 fps and propeller revolutions exceed 2300 rpm.

Although propeller efficiencies in the neighborhood of 85% can be obtained with metal-blade propellers if tip speed and rpm are relatively low, metal-blade propellers have not come into general use because of their increased cost and weight.

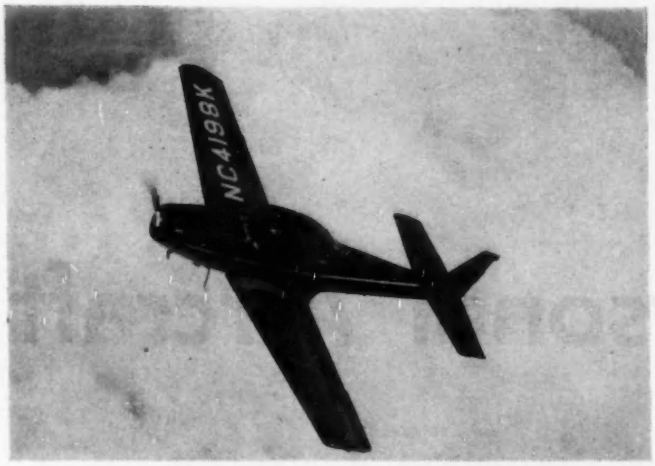
A large percentage of personal aircraft is using flaps. The exceptions are mostly low-priced, 2-place models which omit flaps to keep cost down.

The improved detail design and manufacture of current aircraft, both newly designed and redesigned, reflect the larger engineering staffs, increased test facilities and better production equipment that the war enabled manufacturers to acquire. Drop hammers, hydro presses, stretching presses, automatic riveters, and stack routers have improved forming, handling, and assembly of sheet-metal parts—although they have not made much of a dent in costs.

Synthetic glues resistant to moisture and fungus have improved fabrication of wood parts. Furnace brazing and induction brazing have been developed to the point where they are strictly reliable. Plastics substituted for glass are saving weight and cost. Synthetic-rubber fuel cells are giving less trouble in service than metal tanks.

Representative personal aircraft in current production are shown on the following pages. . . .

* Paper "Personal Aircraft—Problems and Progress" was presented at SAE Annual Meeting, Detroit, Jan. 13, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)



Navion

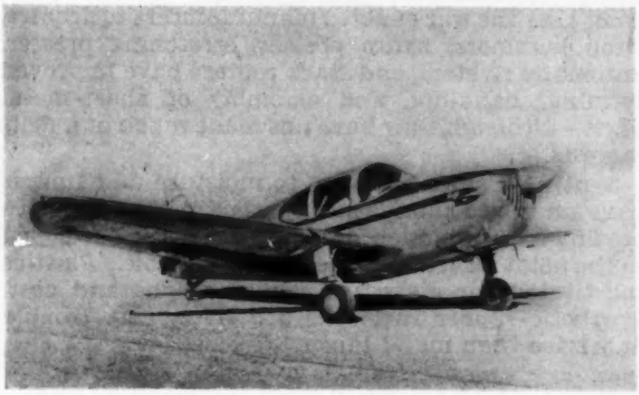
No. Places	4
Engine (Continental), hp	185
Wing Area, sq ft	184
Span, ft	33.3
High Speed, mph	157
Range, miles	500
Gross Weight, lb	2750
Length, ft	27.2
Rate of Climb (SL), fpm	830
Useful Load, lb	1020
Wing Loading, psf	14.6
Service Ceiling, ft	15,600

The Navion is a low-wing, all-metal design with retractable tricycle landing gear. The standard airplane comes equipped with a fairly complete set of instruments, starter, generator, landing light, two-position propeller, and radio. Flying and performance characteristics are excellent. North American Aviation developed the airplane, produced it for a while, then sold the rights to Ryan.

Beech

	"Bonanza"
Engine (Continental), hp	165
Wing Area, sq ft	178
Span, ft	32.8
High Speed, mph	180
Range, miles	600
Gross Weight, lb	2650
Length, ft	25.2
Rate of Climb (SL), fpm	920
Useful Load, lb	990
Wing Loading, psf	14.9
Service Ceiling, ft	16,500
Retail Price, $	10,990

Beech designed the "Bonanza" to fulfil the businessman's requirements. Included in standard equipment are a controllable-pitch propeller, two-way radio, heaters, and a complete extra set of instruments. This all-metal low-wing airplane is characterized by its V-tail. The landing gear incorporates nose-wheel steering and is fully retractable. Tightly fitting doors completely cover the wells when the gear is retracted. All movable control surfaces, including flaps, are fabricated from magnesium sheet. A relatively slow-turning large-diameter propeller maximizes efficiency while minimizing noise. The design received extensive lab and service testing before release to production.



Swift

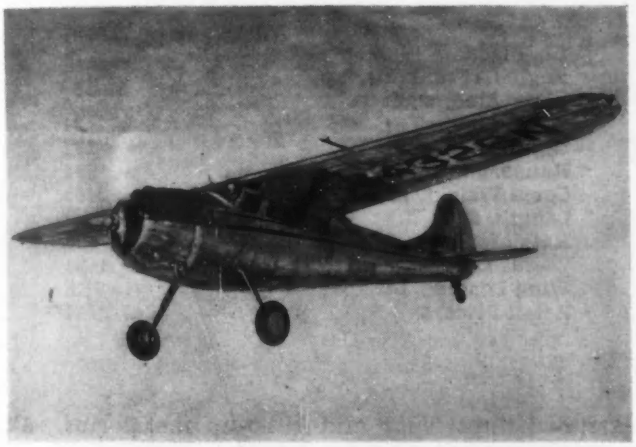
No. Places	2
Engine (Continental), hp	125
Wing Area, sq ft	132
Span, ft	29.3
High Speed, mph	150
Range, miles	425
Gross Weight, lb	1710
Length, ft	20.9
Rate of Climb (SL), fpm	1000
Useful Load, lb	571
Wing Loading, psf	13
Service Ceiling, ft	16,000
Retail Price, $	4495

Texas Engineering and Mfg. Co. produces the "Swift," a low-wing, well streamlined design with retractable landing gear. The airplane is intended for high-cruising-speed operation. It comes equipped with an Aeromatic constant-speed propeller and usual instruments and accessories.

Luscombe

	"Silvaire DeLuxe"	"Silvaire Sedan"
No. Places	2	4
Engine (Continental), hp	90	165
Wing Area, sq ft	140	165
Span, ft	35	38
High Speed, mph	128	140
Range, miles	500	500
Gross Weight, lb	1400	2280
Length, ft	20	23.5
Rate of Climb (SL), fpm	850	900
Useful Load, lb	530	940
Wing Loading, psf	10	13.8
Service Ceiling, ft	17,000	17,000
Retail Price, $	3395	7470

Luscombe offers two all-metal personal airplanes characterized by their high-wing arrangement with single-brace strut to the front spar. Structure is simple, yet aerodynamic form is reasonably good. Interior styling is attractive, and the four-place model is especially roomy.



Cessna

	Model 140	Model 170	Model 195
No. Places	2	4	5
Engine (Continental), hp	90	145	300
Wing Area, sq ft	159	175	218
Span, ft	32.8	36	36.2
High Speed, mph	125	140	180
Range, miles	470	480	750
Gross Weight, lb	1450	2200	3350
Length, ft	21.5	25	21.5
Rate of Climb (SL), fpm	690	690	1200
Useful Load, lb	590	1000	1320
Wing Loading, psf	9.1	12.8	15.4
Service Ceiling, ft	15,600	15,500	18,300
Retail Price, $	3345	5995	14,950

Cessna has three postwar models—all strictly new high-wing designs, basically of all-metal construction, and of good aerodynamic form for fixed landing gear types. Models 140 and 170 have strut-braced, fabric-covered wings; Models 195 has cantilever wings. Spring-type main gear are used. Because Models 140 and 170 have many common or similar parts, the 170 was unusually economical to produce. About 7000 of the two models have been produced.

Ercoupe

No. Places	2
Engine (Continental), hp	85
Wing Area, sq ft	143
Span, ft	30
High Speed, mph	120
Range, miles	450
Gross Weight, lb	1400
Length, ft	20.8
Rate of Climb (SL), fpm	560
Useful Load, lb	586
Wing Loading, psf	9.8
Service Ceiling, ft	11,000
Retail Price, $	4200 (Approx.)

The postwar Ercoupe is a refinement of the prewar model. The low, fabric-covered cantilever wing is rectangular and of constant thickness. The remainder of the structure is all metal. Limitation of rudder and elevator controls prohibits spinning and stalling.

Bellanca

	"Cruisair Senior"
No. Places	4
Engine (Franklin), hp	150
Wing Area, sq ft	162
Span, ft	34.2
High Speed, mph	160
Range, miles	600
Gross Weight, lb	2150
Length, ft	21.3
Rate of Climb (SL), fpm	1130
Useful Load, lb	938
Wing Loading, psf	13.3
Service Ceiling, ft	22,000

The postwar Bellanca "Cruisair Senior" is an improved version of the prewar "Cruisair" with better performance, appearance, and utility. The main gear of a main-gear-plus-tail-wheel landing unit retracts, leaving about one-third of the wheel diameter exposed. The welded steel-tube fuselage is fabric covered. Constant-speed propellers are available.

Stinson

	"Voyager"
No. Places	4
Engine (Franklin), hp	165
Wing Area, sq ft	155
Span, ft	34
High Speed, mph	133
Range, miles	554
Gross Weight, lb	2400
Length, ft	25.2
Rate of Climb (SL), fpm	580
Useful Load, lb	1180
Wing Loading, psf	15.5
Retail Price, $	6444

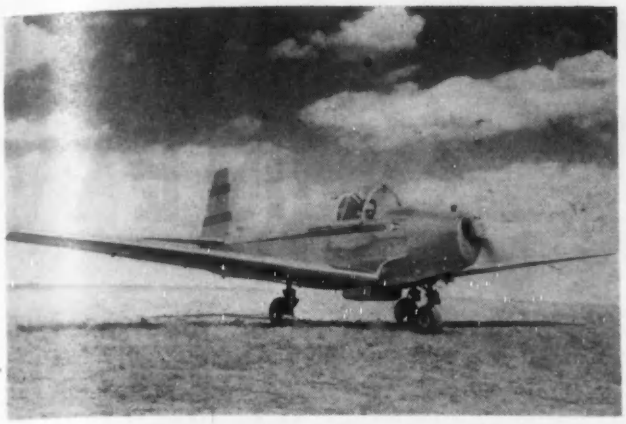
The postwar four-place Stinson "Voyager," developed from the prewar three-place model, features good interior styling, comfort, nice flying qualities, and low price. It is the most popular four-place airplane produced to date.

Fabric covers the welded-steel-tube fuselage and the strut-braced high wing. Within the wing are an extruded dural spar and built-up metal ribs. All movable control surfaces and tail surfaces are all metal. The landing gear incorporates a conventional tail wheel and main gear equipped with hydraulic shock absorbers. Controllable propeller, landing lights, and extra instruments are available at extra cost.

Piper

	"Cub Special"	"Vagabond"	"Family Cruiser"
No. Places	2	2	4
Engine (Continental), hp	90	65	108
Wing Area, sq ft	179	148	180
Span, ft	35.2	29.2	35.5
High Speed, mph	110	100	123
Range, miles	300	250	500
Gross Weight, lb	1220	1150	1850
Length, ft	22.3	18.7	25.2
Rate of Climb (SL), fpm	900	530	575
Useful Load, lb	490	495	850
Wing Loading, psf	6.8	7.8	10.3
Service Ceiling, ft	18,000	10,500	10,500
Retail Price, $	2572	2175	3825

Piper designs are all high-wing, strut-braced monoplanes with welded-steel-tube fuselage, dural spars, and metal ribs with fabric covering. The composite construction facilitates maintenance. A conventional tail-wheel gear is used with shock-cord-sprung main gear. Appearance, detail design, and workmanship of postwar models is considerably improved over prewar models.



Mooney

No. Places	1
Engine (Mooney), hp	25 (at 3900 rpm)
Wing Area, sq ft	95
Span, ft	27
High Speed, mph	100
Range, miles	460 (on 7¾ gal)
Gross Weight, lb	700
Length, ft	18
Rate of Climb (SL), fpm	400
Useful Load, lb	240
Wing Loading, psf	7.4
Service Ceiling, ft	10,500
Retail Price, $	1975

Operating costs of the Mooney M-18 are undoubtedly lower than those of any other airplane being offered now in this country. The engine is the liquid-cooled Crosley "Cobra" modified for use in the airplane and includes dual ignition and 2:1 propeller reduction drive. This reduces propeller rpm to the range for efficient operation.

The forward fuselage section is welded steel tubing covered with sheet aluminum; and the section aft of the cockpit is monocoque plywood. The fabric-covered wings and stabilizer have laminated wood spars with plywood-covered torsion nose section. Movable surfaces have fabric covering over welded-steel-tube frameworks. Besides facilitating maintenance, these construction features afford flexibility in design and tooling.

A simple lever arrangement retracts the landing gear. Shock absorption is provided by rubber compression discs with a friction-snubbed rebound control. The landing gear should require little service.

Aeronca

	"Super Chief"	"Champion"	"Sedan"
Engine (Continental) hp	85	65	145
Wing Area, sq ft	175	170	200
Span, ft	36.1	35.2	37.5
High Speed, mph	100	100	120
Range, miles	370	250	430
Gross Weight, lb	1350	1220	2050
Length, ft	20.4	21.5	25.2
Rate of Climb (SL), fpm	600	500	650
Useful Load, lb	530	470	900
Wing Loading, psf	7.7	7.2	10.25
Service Ceiling, ft	12,500	12,600	12,400
Retail Price, $			4500 (Approx.)

Design simplicity is the keynote of the Aeronca line. All three models have high, strut-braced wings and tail-wheel landing gear. Fuselages are fabric-covered welded-steel-tube structures. "Super Chief" and "Champion" wings have wood spars, metal ribs, and fabric covering. The "Sedan" has all-metal, stressed-skin, constant-chord wings braced by single struts. More than 7000 Champions have been produced. The airplane handles nicely and is easy to service.

For complete coverage of .....

SAE National Transportation Meeting

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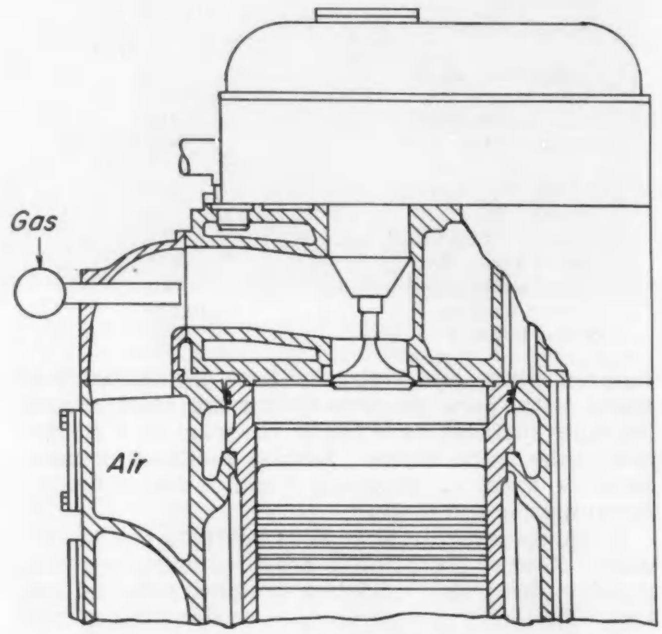

Fig. 1—Method of gas intake for 4-stroke atmospheric gas diesel

NEW

Burn

BASED ON PAPER* BY

Ralph L. Boyer

Vice-President & Chief Engineer
COOPER-BESSEMER CORP.

BOON to users of stationary diesels in areas where gas is available for fuel are the new dual-fuel diesel engines.

They are a form of gas diesel which can burn cheap gaseous fuels with only a small pilot charge of fuel oil or—when the gas supply fails—all fuel oil. Simple controls convert from one fuel to the other immediately and automatically.

Efficiency is good on either fuel throughout the load range. In general, gas is more efficient at full load and oil at low load, on the basis of Btu's consumed.

Already there are enough dual-fuel engines in service to document their low fuel cost, efficient operation, and low installation cost.

When operating on gas, the dual-fuel diesel, like all modern gas diesels, operates on the lean side of stoichiometric gas-air ratio, even at full load. The cylinder contains the gas supply during the compression stroke. Because the mixture is lean, its spontaneous ignition temperature is high and it does not ignite from diesel compression alone. Diesel fuel oil injected into the cylinder ignites spontaneously and fires the gas-air mixture.

For naturally-aspirated 4-stroke-cycle engines, some builders admit the gas to the common air stream, some to the intake-air-valve passages. Some allow the gas to flow into the air stream only while the intake valve is open; some admit gas continuously.

Cooper-Bessemer has found that admitting gas into the main air stream gives uneven distribution. They admit gas to intake-air-valve passages. They found no advantage in timing, so gas is admitted continuously. Fig. 1 shows how it is done.

Gas flows into the passage during two complete revolutions, but flows through the intake valve roughly only one-half of one revolution. Because of the gas accumulation, the first part of the charge is raw gas. Before the arrangement indicated in Fig. 1 was tried, its designers suspected that it would

The dual-fuel diesel engine—which compresses fuel gas and air together, then ignites them with a pilot charge of fuel oil —was scarcely heard of four years ago, but the idea is quite old, Boyer says. In Dr. Rudolph Diesel's U. S. Patent No. 673,160, dated April 30, 1901, Claim I covers "The method of regulating combustion in internal-combustion engines which consists in producing a mixture of air or oxygen and a combustible, compressing the mixture to a temperature lower than the igniting point of the combustible, and introducing under excess of pressure into the mixture a secondary combustible.

* Paper "Status of Dual Fuel Engine Development" was presented at SAE Annual Meeting, Detroit, Jan. 14, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price 25¢ to members, 50¢ to nonmembers.)

DIESELS

Gas or Oil

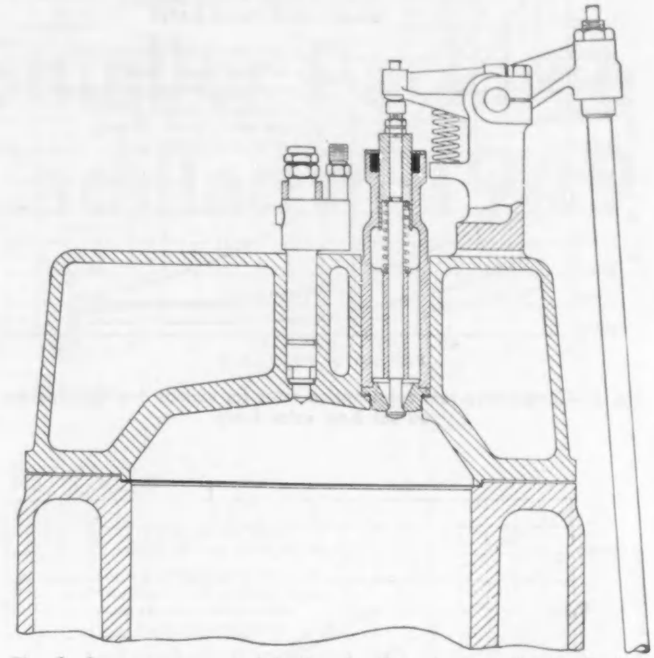

Fig. 2—Low-pressure gas injection for 2-stroke gas diesel. Fuel oil injection is at center of head; gas injection at side

not be conducive to a good mixture. But experiment showed that the arrangement produces the best possible mixture. The inrush of air completely tears apart the gas accumulation.

With 2-stroke-cycle engines, gas is admitted directly to the cylinder under 25 to 50 psi pressure to get the gas in quickly between closing of the exhaust ports and buildup of compression pressure. Fig. 2 illustrates direct gas injection.

Supercharged engines require timing of the gas to prevent its loss during scavenging. Cooper-Bessemer admits the gas to the air stream in 4-stroke engines but finds it necessary to inject the gas directly into the cylinder of 2-stroke engines, although cylinder injection penalizes the quality of the mixture.

Engine controls can be simple, even with dual fuels. If the governor linkage is arranged so that the first half of governor travel controls the gas supply and the second half controls the oil, the engine will switch automatically from one fuel to the other, always burning all the gas that is available. Should the gas supply fail, the governor will drop back into the oil regulating range. When unlimited gas is available, the oil control is manually set to the pilot position.

This control system is enough to satisfy the customer; he wants to burn as near 100% gas as possible when gas is available, and all oil when gas is not available.

Engine builders aim at minimizing the percentage of fuel oil required to pilot gas operation. Currently the pilot charge at any load is the amount of fuel oil needed to furnish 5% of the total Btu consumption at full load. This is even less than the amount needed to keep the engine running without load.

Test results on current production dual-fuel diesels show remarkable efficiency throughout the range of loads, now that early troubles with poor efficiency at low load have been cured. Naturally aspirated 4-stroke Cooper-Bessemer diesels will produce 1 bhp-hr for 6800 Btu on gas at full load and 11,000 Btu at one-quarter load. Fig. 3 shows that this full-load performance is better than that on fuel oil and only 700 Btu higher than fuel oil at one-quarter load.

A supercharged diesel has still better efficiency.

the igniting point of which is equal to or below the temperature due to the compression, substantially as described." As far as is known, Diesel never attempted to use gaseous fuels this way.

Probably the first attempt to compress the mixture of air and gas and ignite it as described by Diesel was made by J. Jones of the National Gas and Oil Engine Co., Ltd. of England in 1939. He concluded that the sensitivity of gas-air mixtures to compression pressures is a matter of gas-air ratio.

Cooper-Bessemer, working independently, came to the same conclusion from experimental work they started soon after.

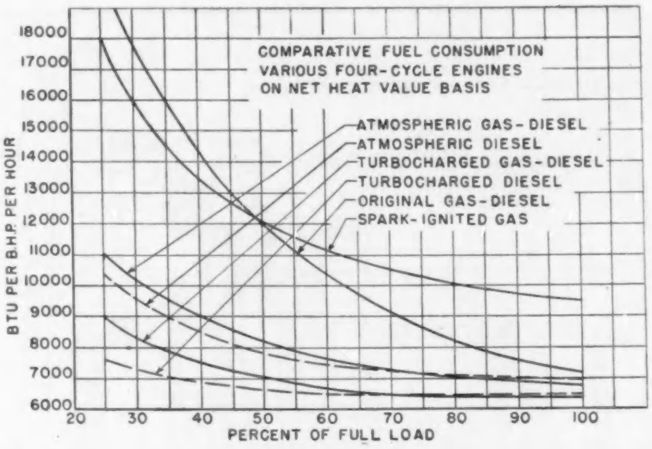

Fig. 3—Comparative fuel consumption rates for various 4-stroke engines on net heat value basis

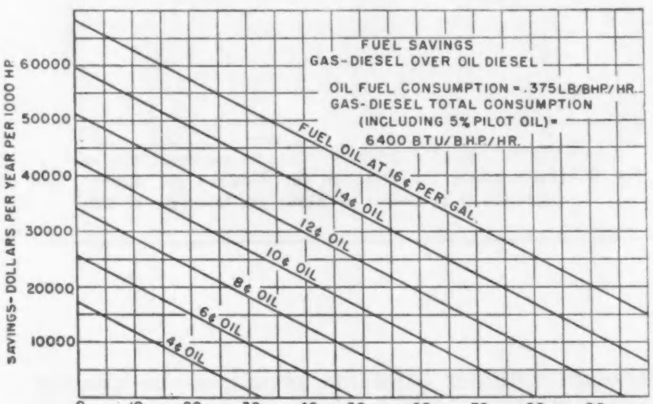

Fig. 4—Fuel savings of gas diesel over oil-burning diesel

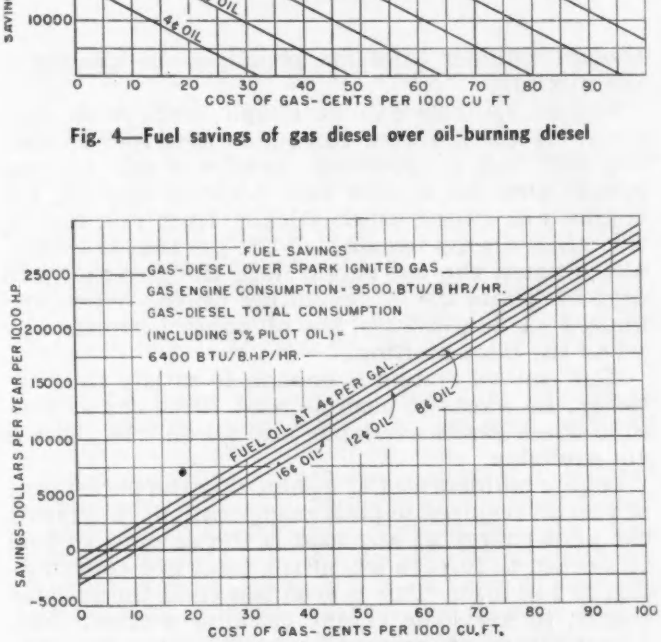

Fig. 5—Fuel savings of gas diesel over spark-ignited gas engine

On gas, it produces 1 bhp-hr at no more than 6400 Btu at full load and 9000 Btu at one-quarter load. At full load, brake thermal efficiency is 40%, which is very close to the ideal thermal efficiency of the cycle. Since there are still traces of raw gas in the exhaust, there is hope that the ideal figure will be approached even more closely.

Service results back up these test results. One installation of an atmospheric 4-stroke engine at the Sulphur Springs Valley Electric Co-operative, Inc. at McNeal, Arizona is rated at that altitude at 975 bhp or 685 kw. Yet it has averaged 6430 Btu per bhp-hr. Part of the reason is that despite its 685-kw rating, the engine has averaged 780 kw. The engine has outperformed its guarantee by a considerable margin. Fuel oil furnished 8.1% of the Btu consumed in 24,409 hr, including several occasions when failure of the gas supply compelled operation on fuel oil.

A supercharged engine generating power for Cherokee, Oklahoma has been consuming less than 6000 Btu per bhp-hr according to their records—and this at a municipal plant where load factors vary widely.

Dual-fuel diesels present no extra service problems. Preignition is unknown. Detonation is practically unknown on 4-stroke engines and never serious on 2-stroke. Although engines are rated at higher bmep's on gas than on oil, cylinder peak pressures and exhaust temperatures tend to be a little lower on gas. Operation is so much smoother with gas that it is easy to tell which fuel is being used by the sound of the engine.

Most dual-fuel engines are using natural gas for their gaseous fuel, but refinery gas, manufactured gas, propane, or even sewage gas can be used.

On the basis of net heat value, natural gas at 65¢ per 1000 cu ft costs no more than diesel fuel at 11¢ per gal. In the gas-producing districts of this country, 20¢ gas is common, 10¢ gas is not unusual, and one large plant in the Southwest is buying gas at 6¢ per 1000 cu ft. Some users get special low rates on gas for their dual-fuel diesels through contracts that allow the gas supplier to cut off the gas when he finds it convenient, usually during periods of cold weather.

Fig. 4 shows that burning 65¢ gas in a gas diesel instead of 14¢ fuel oil in an ordinary diesel saves $25,000 per year per 1000 bhp in the fuel bill. With oil at the same price and gas at 10¢, the saving goes to $54,000.

When the comparison is between gas diesels and gas-burning otto-cycle engines, diesels will save money on fuel wherever gas costs more than 10¢ per 1000 cu ft, Fig. 5 shows. Where gas costs 65¢, diesels will save about $17,000 per year per 1000 bhp. Where gas costs 10¢, the saving is small, but dual-fuel diesels offer the advantage of convertibility.

Dual-fuel diesels offer special advantages in oil field drilling. There, operations can be started on fuel oil, then switched to gas if and when it flows. The gas costs nothing.

Installation costs of dual-fuel diesels, and gas diesels in general, are attractively low. For a given engine size, they give about 75% more power than spark-ignited gas engines. The cooling equipment that took care of a 70-psi bmep spark-ignited engine will serve a gas diesel rated at 120 or 125 bmep because the new engine with its greater thermal efficiency loses less heat to the water jackets. Therefore, the dual-fuel engine requires no more space and no sturdier foundations than competing engines with lower ratings.

More Engine in Smaller Package Achieved by Cadillac in 1949

BASED ON PAPER* BY

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(This paper will be printed in full in SAE Quarterly Transactions)

DESIGN coupled with research gained seven-way improvement for the 1949 Cadillac engine over its 1948 predecessor . . . (1) higher compression ratio, (2) reduced weight and size, (3) cleaner design appearance and good service accessibility, (4) smoother performance, (5) better fuel economy, (6) increased performance, and (7) greater durability.

As is noted in Table 1, which points out most of these differences, compression ratio of the new engine is 7.5 as against 7.25 of the 1948 powerplant. The higher ratio requires 88 Research octane number gasoline for satisfactory knock-free operation at sea level, with standard spark setting. As low as 84 octane number fuel can be used without much performance loss by retarding the spark. The engine also will tolerate high sensitivity fuels because it is free from high-speed detonation.

Basic design of the new engine can be raised to a 12.1 to 1 compression ratio with good results. Rate at which compression ratio of this engine goes up will depend on factors related mostly to the petroleum situation.

Greater compactness of the 1949 engine package is shown in Fig. 1, which is a side-view of the new design superimposed over last year's model. Height to the carburetor flange has been reduced 3 11/16 in. while overall length has been shortened 4 3/32 in.

Weight reduction follows these dimensional decreases. Table 2 summarizes 1948 and 1949 engine

* Paper "The New Cadillac Engine," was presented at SAE National Passenger Car, Body, and Production Meeting, Detroit, March 10, 1949. (This paper is available in full in multilithographed form, from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Table 1—Design Characteristics of 1948 and 1949 Cadillac Engines

	1948	1949
Bore, in.	3 1/2	3 13/16
Stroke, in.	4 1/2	3 5/8
S/B Ratio	1.29	0.95
Displacement, cu in.	346	331
Rod Length, in.	8 3/4	6 5/8
L/R Ratio	3.89	3.66
Crank Centerline to Block Top Dimension, in.	13.22	10.44
Compression Ratio	7.25	7.50
Max. Brake Torque, lb-ft	259*	268*
Max. BMEP, psi	113*	122*
Brake Horsepower	124*	133*
Horsepower per Cubic Inch	0.358	0.402
Engine Weight, lb	887	699
Pounds per Horsepower	7.15	5.25

* Performance data to General Motors Test Code Standards as installed with complete equipment.

Table 2—Weight of 1948 and 1949 Cadillac Engines Compared

	1948	1949	Difference
Total Engine Dry (Standard Transmission)	886.92	699.11	−187.81
Radiator (Core & Tank)	40.00	25.59	− 14.41
Total Engine & Radiator Dry	926.92	724.70	−202.22
Oil	13.30	9.50	− 3.80
Water	52.15	37.55	− 14.60
Total Engine & Radiator Wet	992.37	771.75	−220.62
Weight of Major Engine Parts			
Cylinder Block & Crankcase	292.30	180.20	−112.10
Cylinder Heads	65.70	87.27	+ 21.57
Total Block & Heads	358.00	267.47	− 90.53
Crankshaft	91.18	61.50	− 29.68
Flywheel & Ring Gear	49.66	32.89	− 16.77
Connecting Rod Assembly	18.69	13.36	− 5.33
Intake & Exhaust Manifolds	54.13	44.10	− 10.03
Camshaft	13.28	7.91	− 5.37
Complete Valve Train	47.37	42.63	− 4.74

weights and tabulates principal components which have contributed to weight reduction in the new engine. The 220-lb over-all decrease in engine and radiator installed weight has decreased steering effort considerably because of the car's lighter front-end loading. Most important items in component weight reductions are the cylinder block and crankshaft. The 1949 overhead valve mechanism is 10% lighter than the valve mechanism used in 1948.

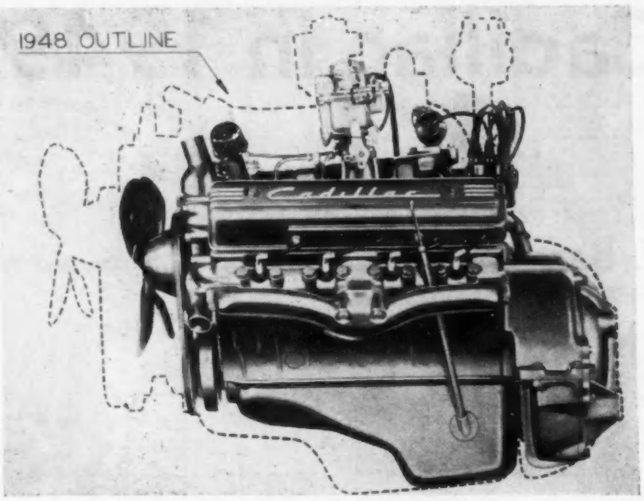


Fig. 1—How Cadillac's 1949 powerplant package compares with the 1948 engine

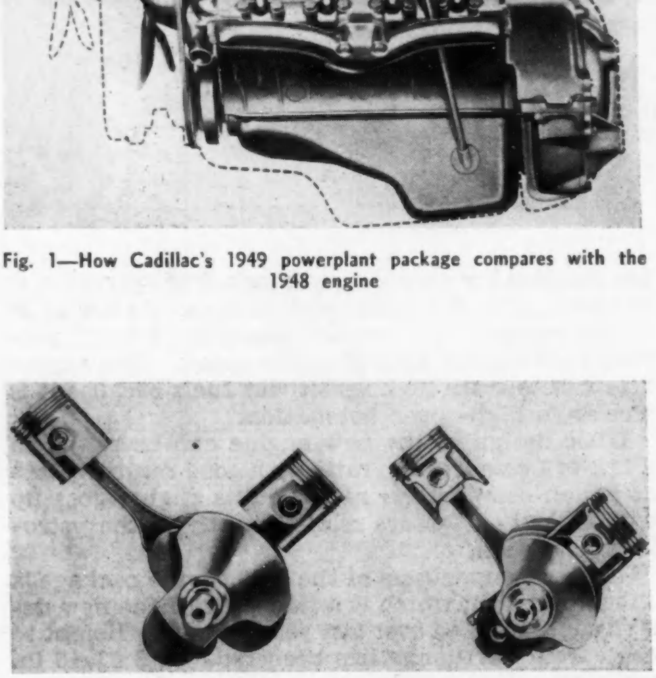


Fig. 2—Crankshaft and piston assembly of the 1948 (left) and 1949 Cadillac engines compared

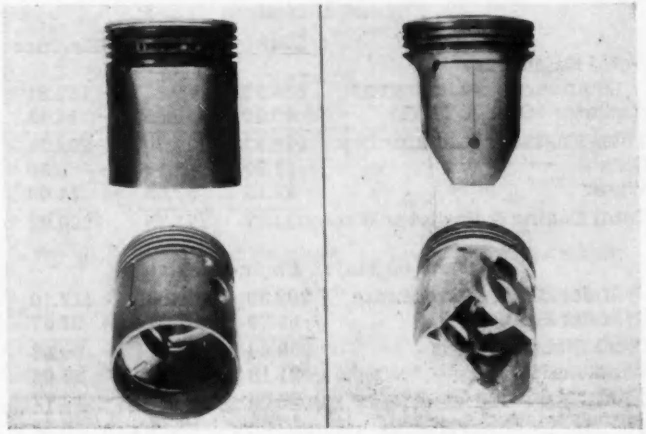


Fig. 3—The slipper piston developed for the new Cadillac engine, at right, is an important factor in design compactness. At left is the full-skirt type used in earlier engines

Fig. 2 compares the 1948 and 1949 crankshaft and piston assemblies, major factors in the weight and size reduction. Notice how the 1949 slipper piston nests between the crankshaft counterweights, permitting a center-of-crank-to-top-of-block dimension of 10.44 in.—2¾ in. less than in 1948.

In addition to making possible a smaller engine package, the slipper-type piston weighs 14% less than the full-skirt type. See Fig. 3 for a comparison of the two designs. The new piston design was reduced to these required basic elements: a head portion to support the combustion load and to carry the rings; extensions from this head for transfer of load to the piston pin; and slipper-type skirts of an area which duplicated the normal contact pattern in a full-skirted, cam ground piston.

The slipper piston construction with the large openings above the piston pin boss allows more splash oil to enter the piston area between the head and connecting rod. This provides the necessary piston pin lubrication and eliminates the need for a gun-drilled rod. Connecting rod bearings also benefit from omission of the gun-drilled hole in the critically loaded area under the rod column.

Fig. 4 depicts the compact arrangement of internal parts.

General design cleanliness and accessory placement are demonstrated by the three-quarter view in Fig. 5. All components are located for good service accessibility. For example, the fuel pump is centrally located, away from the exhaust manifold heat. Carburetor bowl and fuel pump temperatures are 30F lower than in the 1948 engine, improving resistance to vapor lock.

The wedge belt design was selected to drive the

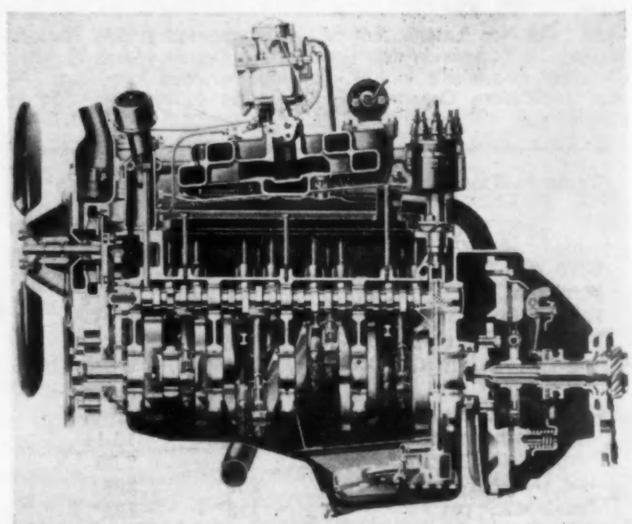


Fig. 4—Shown here is how the internal parts—including crankshaft, bearings, rods, and camshaft—are compactly arranged. In front is the dual water pump with integral bypass from the thermostat housing. Also indicated is the division tract of the intake manifold to show the ribbed sump which traps excess fuel in cold starting and provides ample hot-surface area for quick warmup. Vertical distributor and oil pump drive are at the rear section of the engine

water pump, fan, and generator because of its longer life and greater capacity. The generator is located for easy accessibility in the car, while the distributor is mounted at the rear of the engine—on a common drive with the oil pump—where it gets excellent water-splash protection. The starter also is readily accessible and a very short lead is required from starter to battery.

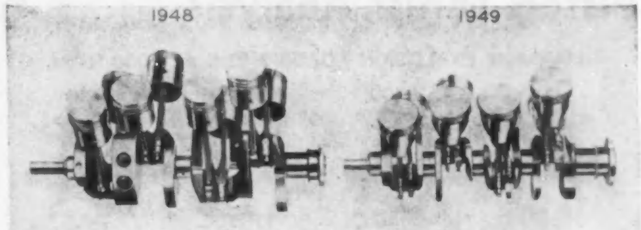
Contributing to smoother operation of the new powerplant are the reductions in crankshaft length and counterweight diameters, portrayed in Fig. 6. Reduction in polar moment of inertia from 1.95 in.-lb per sec^2 in 1948 to 0.93 in 1949 has helped materially to improve the car's accelerating ability. Total rotating inertia, including flywheel, torus cover, and rotating mass of the connecting rods, has been reduced from 5.12 to 3.35 in.-lb per sec^2—a 34% improvement.

Additionally the 1949 crankshaft's primary torsional period has been raised from 15,600 to 19,200 cpm; this stems from the weight reduction combined with increased stiffness due to better overlap of the crankpins and main journals. The tabulation in Fig. 6 shows the relative torsional deflection of these two shafts under maximum power conditions. Note that the objectionable fourth order period of the 1949 engine is above the normal driving range.

Engine roughness observed in our early experimental high-compression engines came primarily from excessive deflection and vibration of structural members. Compactness of the new design made it possible to reduce the weight of main structural members and still provide rigidity to support the loads.

The crankcase, Fig. 7, has been designed to distribute loads evenly throughout the entire structure. Notice the five transverse bulkheads that tie the two blocks into a single rigid unit. A heavy internal rib at the oil pan rail section anchors these bulkheads and they are connected directly to the individual cylinder barrels. Vertical ribs, on the external water jacket surface and also on the surface inside the V, tie the cylinder-head screw bosses into the block structure so that explosion loads on the cylinder heads are carried down to the bulkheads, which also receive the loads from the crankshaft.

Their deep section makes the cylinder heads very rigid. They are attached to the block through five screw bosses adjacent to each cylinder barrel; these members also transfer tension loads to the bulkheads. Uniform spacing of cylinder head screws



TORSIONAL DEFLECTION AMPLITUDE-DEGREES

1948					1949	
WITH BALANCER	BARE SHAFT	RPM	TORSIONAL ORDER	RPM	BARE SHAFT	WITH BALANCER
.08	.16	1950	EIGHTH	2400	.19	.08
.10	.10	3100	FIFTH	3840	.25	.03
.31	.62	3900	FOURTH	4800	ABOVE TEST RANGE	

Fig. 6—Shortening crankshaft length and counterweight diameters of the '49 engine makes for smoother operation

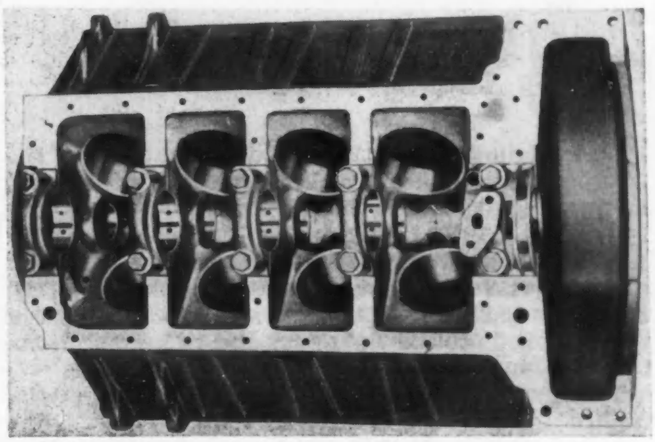


Fig. 7—This bottom view of the crankcase indicates rigidity of the new engine design. The two blocks are joined to act as a single unit by the five transverse bulkheads

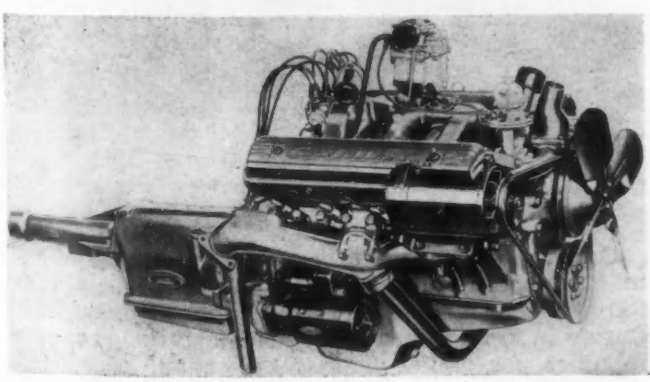


Fig. 5—External appearance of the engine also has been kept clean, with particular emphasis on service accessibility

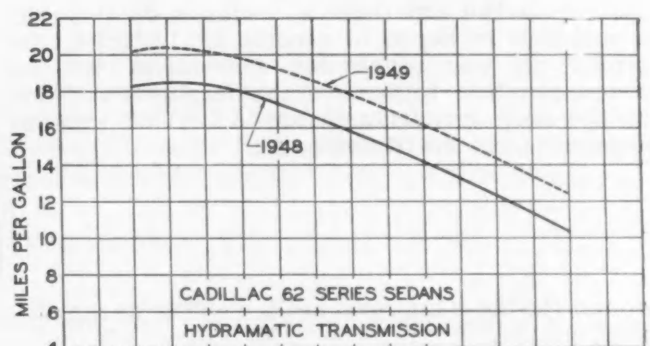
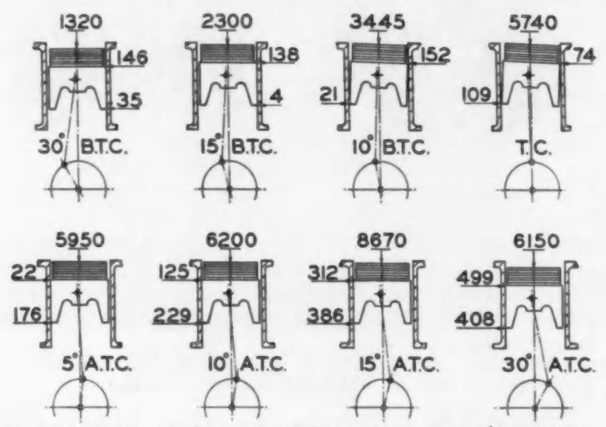


Fig. 8—Road fuel economy of the 1948 and 1949 engines

Designing Durability into

Piston Thrust Analysis

MAXIMUM EXPLOSION LOADING-LBS. AT 2000 R.P.M.



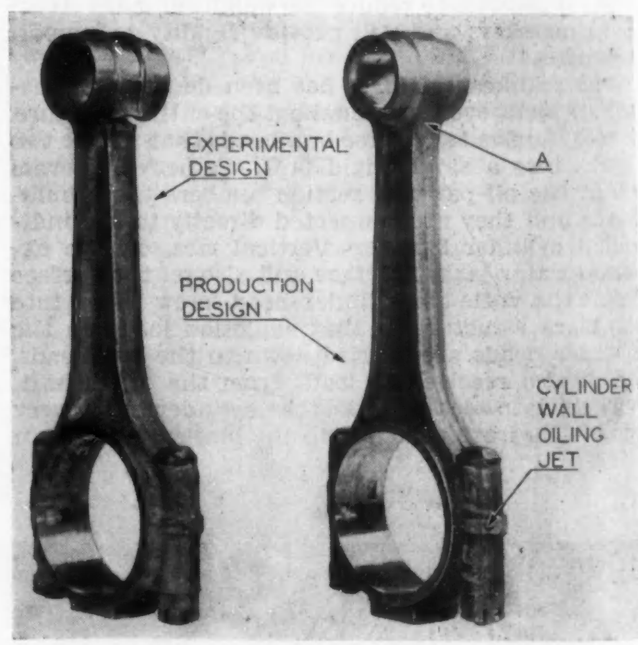
LBS. REACTION AT TOP & BOTTOM OF SKIRT - 1/16 PIN OFFSET

To minimize piston slap and uneven loading of the skirt thrust surfaces during the power cycle, we selected an optimum piston pin offset of 1/16 in. toward the major thrust side.

Tests with the piston pin on center indicated a very high unit thrust loading at the top of the skirt. This coupled with an instant change of the piston from the minor to major thrust side of the bore induced audible piston slap under some conditions. As the explosion load increases near top center, the 1/16-in. offset produces a correcting moment to the piston head which lowers the maximum unit thrust reaction at the top of the skirt.

Note from the pictorial analysis that the piston requires a 20-deg crank motion (from 15 deg BTDC to 5 deg ATDC) to complete a very slow thrust-reversal action. With this construction, the lower or close-clearance edge of the skirt contacts the major thrust side of the bore first, further reducing slap tendency.

Connecting Rods



The original connecting rod design, at left in the photograph, failed at 640,000 cycles. Tension test load was equivalent to the reciprocating load at 4500 rpm plus a 10% safety factor, or 2400 lb. Compression loads were based on maximum explosion pressure at 8 to 1 compression ratio plus the 10% factor, or 10,800 lb. Maximum weight control stock was removed around the upper end to keep sections to a minimum.

The final design, shown at right, yielded only one failure in test—at 8,615,000 cycles. Three main changes produced this improvement. First, the forging was redesigned to give a more gradual section change between the column and wrist pin boss. Second, a constant central section around the piston pin end was developed using hollow milling, concentric with the hole for weight control. (Balance control on the experimental rod was obtained by a variation in depth of central cut.) Third, the profile cut was terminated concentric with the piston pin hole to remove the forging flash line in the most critical area at "A." This removes scratches from the trimming die.

around the bores improves sealing ability of the all-metal gasket.

At 7.5 compression ratio, maximum explosion-load reaction on each combustion chamber of the 1949 engine is 8670 lb, 32% less than the 12,800-lb cylinder head load of the 7.25 compression ratio engine—the 1948 model.

Gains made in fuel economy, the fifth improvement attained in the new design, stem largely from improved mechanical and thermal efficiencies. Curves in Fig. 8 indicate the road-load fuel economy in miles per gallon against car speed, comparing the 1949 and 1948 Model Series 62 cars. Note the increase of up to 20% at 80 mph on these constant

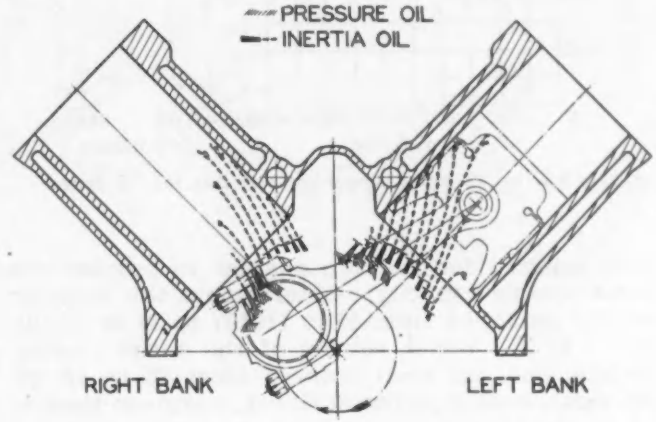
the Cadillac Engine

Cylinder Wall Lubrication

Adequate oil supply on the cylinder walls immediately after cold start is necessary to control cold scuffing and piston slap. Spraying a fine oil stream from the connecting rod at the right time does the trick. Best oil-stream trajectory in a V-type engine can be obtained by designing each rod to oil the corresponding cylinder in the opposite bank.

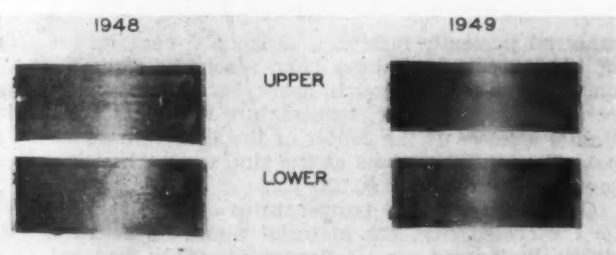
In the new Cadillac engine, most desirable location for the jet is on the split line of the connecting rod. The hole is formed by a milled cut across the face of the cap and a counterbore in the cap around the bolt hole. This jet location avoids stress raisers in the highly-loaded tension section of the rod.

Plotted here is the observed trajectory of 10,000-sus viscosity oil at 10-deg increments of crankshaft angle, at 700 rpm. Note that maximum trajectory height occurs after the jet orifice passes registration with the hole in the crankpin for each cylinder; delivery continues as much as 90 deg past registration. Jets supply ample lubrication on the upper sides of the cylinders which are critical areas, while oil flows to the lower sides via gravity.



COLD STARTING OIL PATTERN AT 700 R.P.M.

Connecting Rod Bearings



131 HOURS AT 4250 R.P.M. | 541 HOURS AT 4250 R.P.M.

4250 R.P.M. BEARING LOADS	1948	1949
MAXIMUM INERTIA LOAD - LBS.	5220	3830
PROJECTED BEARING AREA SQ. IN.	2.49	1.96
UNIT BEARING LOAD P.S.I.	2050	1950
2000 R.P.M. OPEN THROTTLE BEARING LOADS		
NET BEARING LOAD - LBS.	5600	7870
UNIT BEARING LOAD P.S.I.	2250	4000

Life of high-speed connecting rod bearings has been extended because of the lower inertia forces and other design features of the 1949 engine. In the photograph, the pair at right illustrate typical 1949 bearing shells after 541 hr open-throttle testing at 4250 rpm. At left are shown 1948 bearings after 131 hr operation in that engine under the same test conditions.

The tabulation shows the reduction in inertia load due to lighter parts and shorter stroke. Note the high upper-half unit bearing load at 2000 rpm on the 1949 engine.

Although Durex Type 100A bearings shells are used in both engines, in the 1949 design the oil holes for cylinder wall and piston pin lubrication have been eliminated from the upper shell. It is a major factor in extending bearing life in the new engine. This is so because of the noninterrupted oil film which exists under the rod column at the instant of maximum explosion load application.

speed curves. Overall gain in driving economy averages about 2 mpg.

At 4000 rpm, engine friction has been reduced from 70 to 54 hp, or 23%. This bears significantly on the greater fuel economy in the higher speed range.

Fig. 9 compares heat rejection of the 1948 and 1949 engines. The cooling system shrunk because of the reduced "boiler surface" of the combustion chamber, cylinder wall, and exhaust ports and also because of lower frictional losses. At 4000 rpm, heat losses to the coolant were lowered from 6500 to 5200 Btu per min—a 20% improvement.

Sketches at right in Fig. 9 show the sizes of radia-

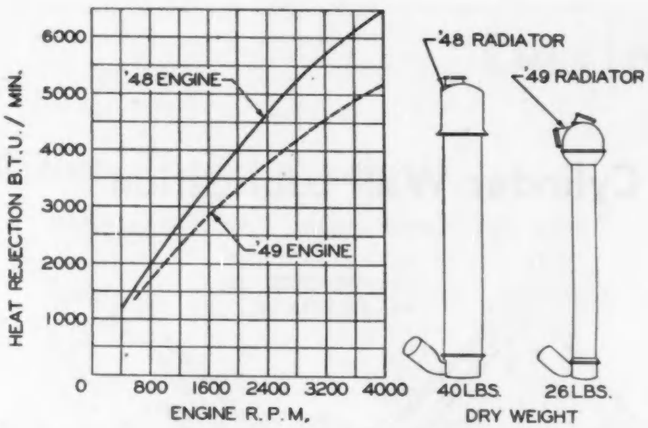


Fig. 9—The '49 engine rejects less heat than the '48 design

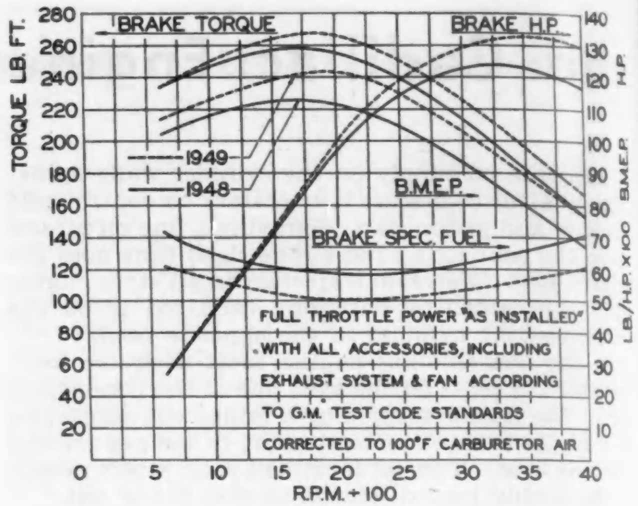


Fig. 10—Full-throttle power of the 1948 and 1949 engines

tors required for the two engines to provide the same cooling capacity. Observe that the radiator weight reduction from 40 lb (1948) to 26 lb (1949) saves 14 lb. Water volume of the entire cooling system also has been reduced from 26 to 18 qt. Although heat rejection is lower, warm-up time is about the same due to reduced mass of engine and coolant.

Design gain No. 6—increased performance—manifests itself in Fig. 10, which indicates open-throttle "as installed" power comparisons between the 1948 and 1949 engines with complete car equipment, including fan and exhaust system. These tests were run according to General Motors test code conditions, corrected to 100F carburetor air temperature.

Note the 8% gain in maximum bmep and the 7% gain in peak bhp. Additionally, the horsepower of the 1949 engine remains higher after the peak. The 14% gain in specific fuel economy throughout the entire engine speed range indicates the improvement in open-throttle mechanical and thermal efficiency.

Best way to demonstrate the improvement in quality and durability of the new engine's components is to point out how these problems were licked in the development work. Four of these projects are discussed on the previous pages.

Severe Cooling May Impair Disc Stress-Strength Ratio

Based on paper by **S. S. MANSON** NACA

(This paper will be printed in full in SAE Quarterly Transactions.)

SEVERE cooling of gas turbine discs can do more harm than good, in some cases.

The reason is that, although strength of the disc material depends on temperature, the stress induced in the disc by the heat of operation depends on temperature gradient. By increasing temperature gradient, cooling may increase stress more than it increases strength.

NACA has been studying this problem. In an investigation of a particular turbine disc with provision for cooling one face, they installed thermocouples on both faces of the disc. Then the disc was subjected to a typical sequence of operating temperatures from starting, through idling and taxiing, to 15 min at rated take-off conditions.

Stress was calculated for the temperature distribution obtained for each face. It was assumed that stresses calculated for the temperatures along the cooled side would be representative of stresses if both sides of the disc were cooled, and similarly, that the stresses calculated for the uncooled side would be representative of stresses if neither side were cooled.

Stresses on the cooled side ran very high. Were the material perfectly elastic, a tangential compressive stress of 120,000 psi would have been reached. Stresses on the uncooled side were much lower because, although temperatures were higher, temperature gradients were lower. Radial stresses at the center of the disc were about 45,000 psi. Tangential stresses at the rim, where they were most severe, ranged about 50,000 psi.

Of course, with the temperatures and stresses indicated on the cooled side, the material would have exceeded its elastic limit, and plastic flow would have occurred. For points along the radius of the cooled face, radial and tangential stresses were translated into equivalent uniaxial stress and compared with elastic-limit stresses corresponding to the various temperatures. The comparison showed that plastic flow in the rim region would have reduced stresses there and at smaller diameters also.

While plastic flow at operating temperatures results in reduced stresses at operating temperatures, it may induce undesirable residual stresses on cooling if rims are continuous, as they are with welded-in blades. (Fir-tree fastenings involve a discontinuous rim and allow rim segments to draw away from blades on cooling.) Calculations showed that tangential tensile rim stresses would reach about 60,000 psi on cooling of the disc used in the experiment. Repeated heating to operating temperatures and cooling to ambient temperature could lead to fatigue cracks at the bases of welded blades, where rim stress concentrations are high. In fact, it may explain the rim cracks already experienced in turbines of welded-blade construction. (Paper "Stress Investigations in Gas Turbine Discs and Blades" was presented at SAE Annual Meeting, Detroit, Jan. 14, 1948.)

Practical Design Pointers On Vehicle Prop Shafts

BASED ON PAPER* BY

George E. Dunn

Chief Engineer,
Universal Products Co., Inc.

IMPOSSIBILITY of achieving perfection in propeller shaft design doesn't nullify the feasibility of good performance by observing proved design practices.

Propeller-shaft design dogma urges, for example, that geometry of the driving system should enforce equal joint angles under all operating conditions. It also advises that critical speed of the shaft and natural frequency of its supports should well exceed maximum required speed.

Since critical speed of shaft depends on its supports, the shaft should be accurately mounted on vehicle parts that are stiff enough and have minimum overhang. Attention to design and fabrication of the shaft itself, particularly to balancing, also stands to better performance. Another longevity-producing practice advised is use of strong enough universal joints that do most of their work at about a 3-deg angle.

Reason for a system geometry in which the universal joints make equal angles between input and output shafts is to get constant speed through the driving system. Two usual ways of achieving this are shown in Fig. 1. Note that in each case the yoke bushings at each end of the intermediate shaft are

* Paper "Universal Joints and Propeller Shafts," was presented at SAE Annual Meeting, Detroit, Jan. 12, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

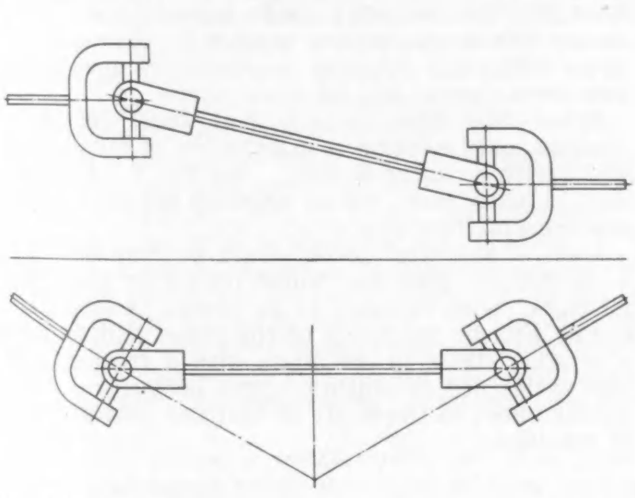


Fig. 1—Two methods of getting constant speed in the vehicle drive system

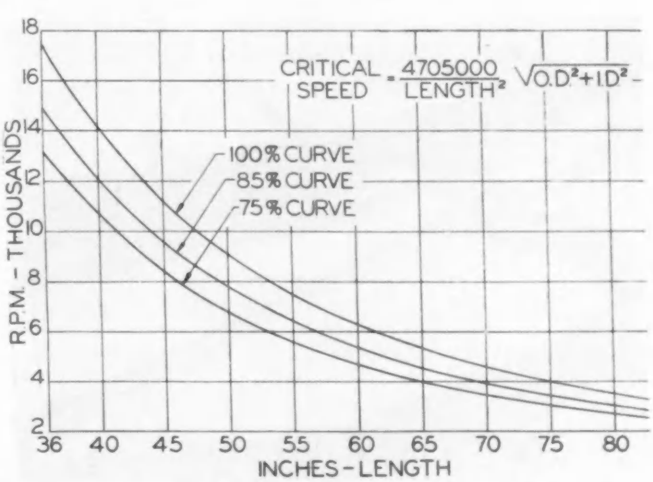


Fig. 2—This critical speed chart for 3½-in. 14-gage tubing is typical of those for various propeller shaft sizes. The designer is strongly urged not to exceed values on the 85% curve

in line with one another, and that there equal angles between input and output shafts.

Centerline of the shafts lies in the same plane in a projection of 90 deg from the views shown. With this arrangement, varying speed in one joint is cancelled out by compensating motion in the other. This will produce constant angular velocity and constant torque transmission in the output shaft.

Despite the two universal joints, this ideal setup does not exist with the Hotchkiss drive construction. Axle windup and other factors in the system cause considerable deviation from the ideal. It is common practice to try to hold angles of the two joints as much alike as possible in the normal operating range of the vehicle.

With only one joint, as in a torque tube drive, variance in universal joint motion must be taken up by elasticity in the driving train.

Any vibration in the vehicle brings a look of sour suspicion directed at the propeller shaft. While the shaft may be vibrating, it is equally possible that the fault does not lie in the shaft at all.

As a matter of fact, there is no inherent vibration in a propeller shaft. It will whip or whirl, but it will not vibrate. There are no stress reversals in the fibers of the shaft when it whips. Any original unbalance is a source of energy-exciting vibration which will be produced in the shaft supports. This applies only to vibration due to shaft unbalance excitation. Forced vibrations from other sources can produce true transverse vibrations in the shaft.

Second fact of life the drive system designer must cope with is that any shaft rotating from zero to high speed passes through certain speeds where instability occurs. This is a reality despite any efforts to balance the shaft. If the shaft is accurately balanced and it passes through these critical speeds quickly, the shaft will settle down and run well again. But the shaft will fail if critical speed is maintained.

Critical shaft speed depends on character of the supports. Heldt[1] makes this clear when he says: "A motor vehicle propeller shaft when running at high speed, will whirl in the same way as a heavy rope which is being swung by two persons. If they cease their whirling effort, their hands will, nevertheless, be carried around in a circle, and so with the propeller shaft supports. The latter consist of universal joints which are fitted to shafts overhanging their bearings. Under the influence of centrifugal force on the propeller shaft, these short shafts will flex in the same plane as the propeller shaft, thus virtually increasing the distance between supports. The effect, of course, depends on the relative stiffness and on the overhang of the connected shafts."

Fig. 2, a typical critical speed chart, brings out this character-of-support influence. Although this is for a 3 ½-in. diameter 14-gage tubing, the same general curve holds for all shaft sizes.

Top curve shows the critical speed calculated from the formula shown, based on the beam theory. The bottom curve shows safe speeds, 75% of critical speed. Values on this curve will be safe for ordinary drive trains. With adequate bearing supports and moderate forced vibrations, the design can approach the center curve—85% of the top one. Going above this curve in any propeller shaft application is generally unwise. Even the most carefully balanced shaft may vibrate when above the 85% figure, and not necessarily from error in the shaft, but again due to character of its supports.

Potentially hazardous is the situation where the shaft is required to run at varying speeds and its maximum speed includes the critical range. There is no assurance that an operator will not maintain these speeds long enough to fail the shaft.

Shaft-Making Complexities

Trend toward longer and larger-diameter shafts in recent years is another factor posing many problems for the propeller shaft manufacturer. The longer the shaft and the greater its diameter, the more difficult it becomes to produce a good shaft, free from general and localized errors.

When the limit to practical shaft length is reached, only solution is installation of a two-piece shaft with a center bearing. See Fig. 3. Although used in many cases, center bearings are not without problems of their own.

Even if universal joints could be manufactured to perfection, they still would offend by producing vibration when running at an angle. A secondary couple acts in the plane of the yokes and imparts a bending stress to the shaft with a frequency of two cycles per revolution. This produces a resonating effect in the shaft at half the critical speed of rotation.

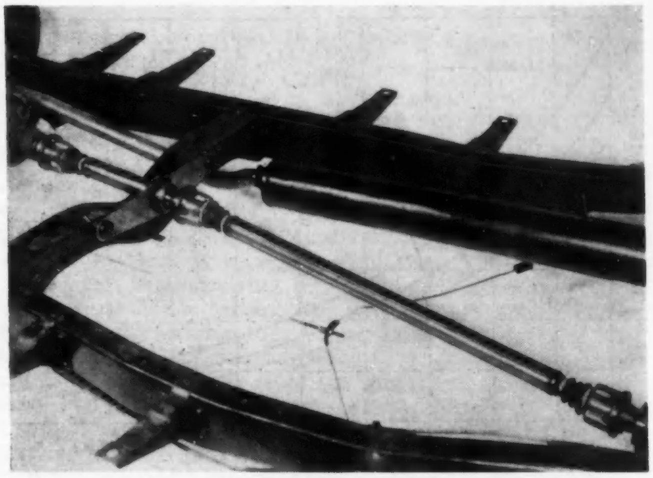


Fig. 3—If a propeller shaft gets too long, it is advisable to use a two-piece installation with a center bearing, as shown

[1] See "The Automotive Chassis," by P. M. Heldt, published by P. M. Heldt, Nyack, N. Y., 1945.

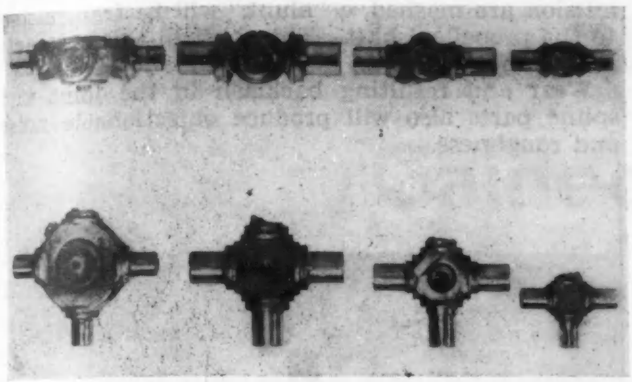


Fig. 4—These typical universal joint field failures all follow the same pattern. Failure begins at the trunnion base, close to where the radius and trunnion meet

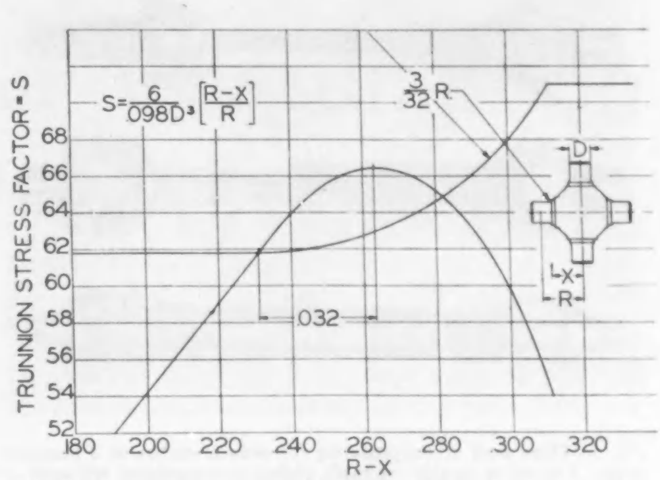


Fig. 5—Trunnion stress curve. Note that maximum stress occurs where failures in Fig. 4 were experienced

If shafts do not run smoothly to their maximum required speed, balancing is necessary. Balancing is at best a compromise. A propeller shaft changes its shape with speed of rotation and there are no intermediate bearings—as in a crankshaft—to help the shaft retain its shape.

A shaft balanced to 1 in.-oz at 4000 rpm may have a 16 in.-oz unbalanced at 1000 rpm, or 64 in.-oz at 500 rpm, and the unbalance forces will be the same in each case. Assume that a given propeller shaft will bend between 500 and 4000 rpm so that its balance changes to the extent of 63 in.-oz. Let it be balanced to 1 in.-oz at 500 rpm. Then the unbalance at 4000 rpm will be 64 in.-oz. The forces will be less than ½ lb at 500 rpm; but since the forces increase as the square of the speed, at 4000 rpm they will exceed 1800 lb.

For this reason shafts must be balanced at high speeds. There are no two speeds—from zero to critical speed—at which the unbalance will be the same. Amplitude of vibration increases with speed until the critical speed is reached. Forces due to unbalance increase with speed, as was shown in the example, and a lesser unbalance must be obtained at high speed than can be tolerated at low speed.

A shaft that shows up well in the balancing machine may perform poorly in the vehicle. In such a case it generally will be found that components on which the shaft is mounted are at fault. If a number of shafts are installed in the same vehicle, often the one giving the best performance shows up poorly in the balancing machine. Here the errors in the shaft and those in the vehicle cancelled each other. We are right back to the same old saw—shaft performance depends on character of its supports.

Because propeller shafts rotate at fairly high speeds, universal joints should be as small and as light as possible and have capacity for satisfactory life.

Practically all modern automotive joints use rollers or needles in their assembly bearings because of the high loadings on the bearings. Because of the characteristics of universal joint motion, the bearings oscillate and do not make complete revolutions. For this reason the joint must run at sufficient angularity to circulate the rollers. At zero angularity there would be no roller motion and true Brinelling would occur. High angularity would shorten bearing life. Optimum angularity is about 3 deg.

Both field and experience and laboratory tests show that breakdown of bearing components follow lubrication breakdown. This points up the need for good sealing and adequate lubrication. While all bearing parts will show wear, greatest damage usually will be found on the cross trunnion. This is to be expected since greatest loads are concentrated on the surface of this part.

Overloading universal joints usually produces failure at the cross trunnion. This failure generally will follow a breakdown of the bearing. Trunnion failures stem almost entirely from fatigue. Examination of failed specimens will show the progression of a crack from the surface to the point where final rupture occurs.

Fig. 4 depicts field failures in different-size crosses made by various manufacturers. Note that failure starts on the radius at the trunnion base, at a point close to the blend of the radius with the trunnion.

The stress curve in Fig. 5 shows that failure at this point is to be expected. The trunnion is a cantilever beam and it is assumed that the center of pressure is at the midpoint of the roller. The stress curve shows a maximum at the point where we saw failures. Stress raisers could cause failures at other points also.

Obviously the trunnion is subjected to the worst possible condition for endurance life—complete stress reversal. In addition to torque reversal in the vehicle, varying angles of the joint while under torque produce changes in directions of stress.

Since laboratory setups cannot duplicate field conditions, experience must largely guide the designer in allowable stress in a universal joint cross. Computed fiber stress in the trunnion should not exceed 64,000 psi using maximum calculated vehicle torque.

Another condition shaft design must provide for

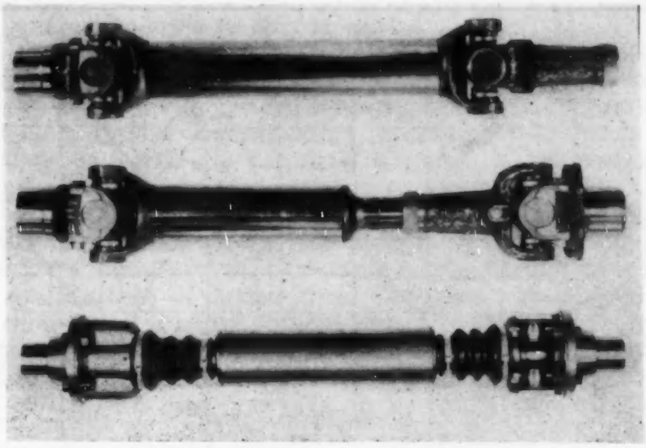


Fig. 6—Three ways of compensating for endwise motion in a propeller shaft. Top, spline outside the shaft, sliding on transmission tail shaft or center bearing shaft. Center, spline within the shaft. Bottom, balls in the joint body raceways

is endwise motion to compensate for lengthening or shortening of shaft, due to relative motion of various vehicle parts.

The three ways of getting the required motion, shown in Fig. 6, are:

1. The spline is outside the shaft proper and slides on the transmission tail shaft, or on a center bearing shaft.
2. The spline is contained in the shaft itself.
3. No spline is used, endwise motion being derived in the joints by balls rolling in the joint body raceways.

Splines must be made accurately. No matter where they are located, they tend to excite vibration and to lower the shaft's critical speed because of necessary clearances and developed thrust loads. The ball and trunnion joint minimizes these effects because of the rolling instead of sliding friction .

Involute splines have some advantage over those with square-sided teeth. They may be generated to close limits and readily lend themselves to shaving, which gives them a final degree of accuracy. Generally they are heat-treated to a hardness that permits this final shaving operation. Involute splines are designed with sufficient bearing area so that unit pressure imposed is within limits permitted by the hardness. When wear develops, they have a beneficial self-centering tendency.

Square-sided splines generally are heat-treated to higher hardness than the involute type. Because of this higher hardness, they may be worked at higher unit pressures. For this reason square-sided splines are used extensively for heavy-duty applications. They must be ground all over for high accuracy.

Last item to be noted about propeller shafts is that the noises which emanate from them all are caused by vibration. Sources of vibration vary, but they may be related. Large diameter tubes will produce noise from the high surface speed of the tube when running with practically no vibration.

When a vehicle is started under a high torque load, a shaft equipped with a spline will sometimes give rise to a noise known as "spline grunt." It comes from a combination of spline friction and windup in the vehicle parts. In some cases when the vehicle is stationary and gears in the transmission are meshed, a "klunk" will be transmitted to the propeller shaft. The shaft in this case acts as a sounding board.

Wear and resulting backlash in the joint and spline parts also will produce objectionable noise and roughness.

■

On the other hand . . .

Says Discusser

R. R. BURKHALTER

Spicer Mfg.,
Division of Dana Corp.

The author's statement that the cross trunnion is subjected to the worst possible condition for endurance life (complete stress reversal) is questionable; full reversal is obtained only in shifting from forward to reverse and a partial reversal during deceleration. Possibly the cross trunnion failures are the result of a sustained high stress aggravated by the variation caused by torsional vibration.

The controversy of involute splines versus straight-sided splines for the universal joint telescoping members seems to be a matter of individual preference or convenience in manufacture. Most universal joint manufacturers, including ourselves, are currently using both types.

The self-centering feature of the involute spline does not seem to offer any advantage in the sliding spline position since the spline must be centered with and without a torque load. In fact, most vehicles are more sensitive to universal joint spline clearance during free rotation (when the vehicle overruns the engine) than under heavy torque loads.

To get the best centering possible, most sliding splines use a piloting fit on the outside diameter as this surface is not subject to torque loads and wear is negligible. The straight-sided spline has the advantage of increased area of contact.

Optimum universal joint angle is specified as 3 deg to circulate the rollers. This angle may be desirable for fixed-angle installations; but in vehicles where the propeller shaft is connected to a flexible mounted axle, the normal spring movement provides sufficient roller circulation. The normal operating range is between an empty and a fully-loaded vehicle. To maintain minimum universal joint operating angles, the drive line should be approximately a straight line in the mean position.

High Air Consumption Gained in Waukesha's Comet Diesels

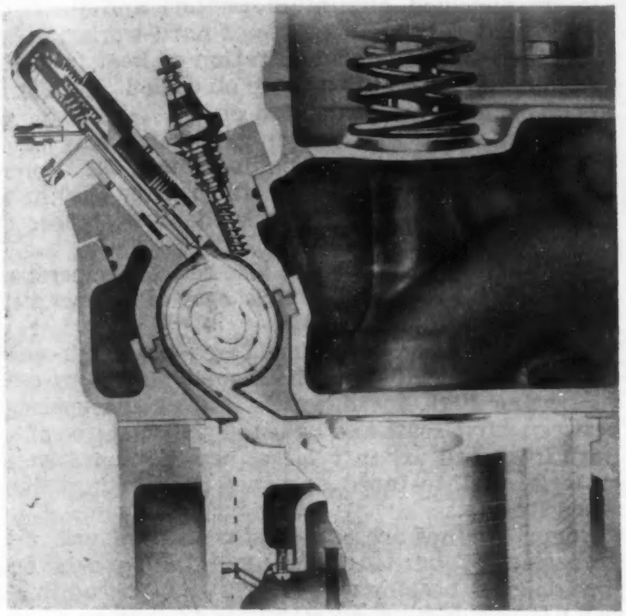


Fig. 1—Two cavities in the piston crown of the Comet diesel combustion chamber aid in burning air over the piston

BASED ON PAPER* BY **J. B. Fisher**
Vice-President and Chief Engineer,
WAUKESHA MOTOR CO.

WAUKESHA'S Mark III Comet combustion chamber gives greater fuel economy and cleaner exhaust because it burns more air. Other design features of this diesel line help minimize maintenance.

Two depressions in the piston crown, connected by a passage leading to the throat of the swirl chamber, help burn more of the air over the piston. See Fig. 1. Ignition takes place in the highly-heated air within the swirl chamber, as in earlier types. The still-burning droplets leave this chamber at high speed (due to the pressure rise in the chamber), traveling down the connecting passage until they meet the sharp promontory where the two depressions meet.

They are then split into two streams of air and fuel which swirl around the depressions in counter-rotation. In this way the fuel droplets meet all the air needed to complete their combustion. To do this we do not need as great a swirl velocity as in the cup itself. Of course, no swirl takes place in the piston depressions until after combustion has started and convection heat losses are much reduced.

"What determines the relative volume of the swirl chamber and that portion of the chamber over the pistons?" you may ask. It would be desirable to use the smallest possible swirl chamber to initiate combustion and to reduce the delay period, completing as much as possible of the combustion over the piston.

We have not been able to reduce the chamber to much less than half the total combustion chamber volume. Any further reduction in swirl chamber capacity can be achieved only at the cost of still higher intensity of air swirl. To use Ricardo's description, "We lose on the swings all that we gain on the round-abouts."

With a very small chamber of less than half the total volume there is danger of over-penetration of the fuel jet; however very small chambers are used successfully in small bore Comets. At Waukesha we are building Mark III Comet engines from 3¼ to 8½-in. bore. By using a 50-50 proportion between swirl chamber and piston cavities, we are able to burn with a clean exhaust as much as 88 to 90% of the total air. Thus we gain about 10% in power over the Mark II performance with the same gain in fuel economy.

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* Paper "Development of a Combustion Chamber for Medium and High Speed Diesel Engines," was presented at SAE Annual Meeting, Detroit, Jan. 14, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.) Announcement of Mr. Fisher's retirement, after 35 years of active service, appeared in SAE Journal last month, p. 68.

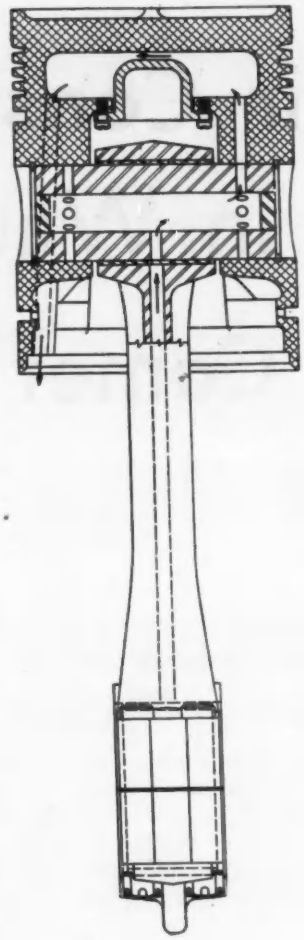


Fig. 2—Oilcooled aluminum pistons withstand heat from the swirl chamber

Reason for this is use of the air swirl to keep feeding air to the burning droplets, rather than letting them be ejected into the space over the piston and depending on chance for this to happen.

Minimum fuel consumption of the Mark III is 0.38 lb per bhp hr, or about midway between the Mark II and the best of the open chamber engines. Power at the clean exhaust limit — usually between 110 and 115 psi bmep—is well above that of the open chamber diesels. It is maintained up to much higher speeds by the compensating action of a heated cup.

Mark III Comet engines will idle for 2 to 3 hr with no tendency to smoke or foul the combustion chamber. It seems that oil field mechanics have an unwritten rule, inherited from the steam engine days, never to shut down an engine from the time they come on duty until they leave. Best testimonial to the clean and complete combustion of these engines is this fact: You can spend day after day testing them in a dynamometer test cell, under all sorts of conditions, with no more discomfort than when working with well-adjusted gasoline engines.

Some diesel engines, when supercharged, can handle a lower grade fuel due to the generally higher temperature of the entire cycle. Ability of the Mark III chamber to digest low-grade fuels is enhanced by these three additional features: (1) use of a large, single-hole nozzle, (2) the very hot lower half of the swirl chamber, and (3) the constant effort made throughout the combustion cycle to feed fresh air to the burning fuel droplets.

To demonstrate this, six fuels were tested in a $5\frac{1}{2}$-in. × $5\frac{1}{2}$-in. Mark III engine. They had widely different properties, particularly in the high percentage of residue in the Conradson test. When supercharged to 10 lb, the engine burned this wide variety of fuel cleanly and with very similar fuel consumption figures. The worst of these fuels tested develops very nearly the same bmep (at the point of just-visible exhaust limit) as the best one, a light gas oil.

We speak of high thermal efficiencies in diesel engines, but seldom tie this in with an equally important item—fuel cost per gallon. It is not wise to develop a diesel engine with even 40% brake thermal efficiency if it requires a fuel so costly that it brings overall cost in line with a competitive engine, burning cheaper fuel, with a 30% brake thermal efficiency.

Other features of the Comet engine worth noting are its oilcooled aluminum pistons and judicious use of materials for long life of hard-worked parts.

To withstand the concentration of heat from the swirl chamber, the pistons are oil cooled. See Fig. 2. About 0.8 gpm are circulated on a $6\frac{1}{4}$-in. piston at 1500 rpm. For this reason the rings stay clean and free, and the sharp point—where the depressions intersect—remains as sharp after 5000 hr of service as when new; this evidences absence of erosion from heat.

The lower half of the cup also resists temperature fluctuations excellently. It is cast of Haynes stellite by the lost wax process.

Head cracking, often suffered by high-speed diesels, is prevented in the Comet engine by concentrating high-speed water jets at the openings between valve seats and by using a special iron alloy. Both inlet and exhaust valves are of exhaust valve material, stellite-faced, and they operate on stellite-faced inserts.

Comet engines are equipped with hardened cast-iron sleeves, hardened crankshaft journals, and steel-backed bearings with F-77 copper-lead lining and 0.001 in. indium overlay.

The crankshaft used in the larger engines is of cast graphitic steels, with journals hardened to 60 Scleroscope. Shaft cheeks are severely relieved adjacent to the journal bearings. This reduces high stress concentration usually found at the fillets and distributes bending loads more evenly over the cheek. Because of the cored openings in the journal bearings, the shaft, shown in Fig. 3, probably can be heat-treated more uniformly than a solid forged one. Its physical properties are comparable to a heat-treated forged shaft of SAE 1045 steel.

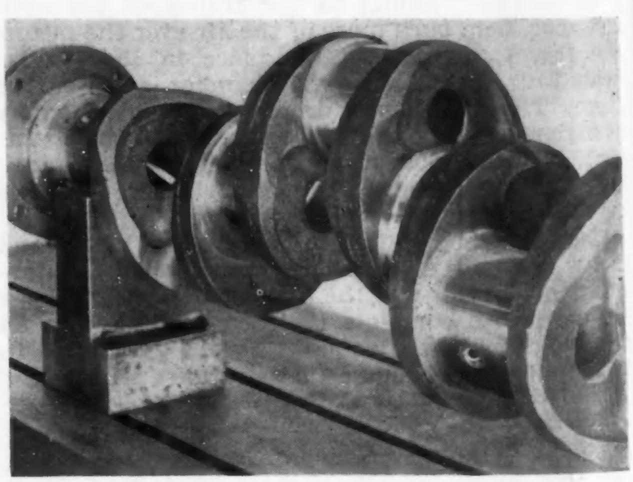


Fig. 3—Larger Comet engines are built with cast graphitic-steel crankshafts which have physicals not unlike heat-treated forged shafts of SAE 1045 steel

MONOCOQUE TRAILERS

BASED ON PAPER* BY

Keith W. Tantlinger

Chief Engineer
Brown Industries, Inc.

PARADOXICALLY, it is possible to increase strength in a monocoque structure by eliminating weight.

Military and commercial aircraft have been built for years with this type of construction for fuselages and wings, and the durability, strength, and light weight of stressed skin design has been well established.

Application of monocoque construction to highway van trailers creates, in effect, a tremendous tube 8 ft wide and 8 ft deep with every inch of its periphery easily carrying a small, but important, part of the total load Fig. 1.

Torsional rigidity is exceptionally high. This is of the utmost importance when entire payloads are hung from the trailer roof—as is the case when transporting halves of beef.

This continuous structural membrane surrounding a valuable cargo is an evident advantage. Highway accidents with fully loaded monocoque van trailers have proven that frequently nothing more than appearance of the outside of the vehicle is damaged. It is not uncommon for loads of sacked grain or flour to survive accidents severe enough to disable the tractor without breaking a single sack.

The best natural analogy for monocoque is an egg shell. It has been said that an egg shell cannot be crushed longitudinally in a person's bare hand.

Stress distribution in a properly designed monocoque structure is so complete that unit stresses are extremely low. Actual electrical strain gage measurements show in all cases less than 6000 psi. Hence material for the structure need not be selected on the basis of tensile strength alone.

Resistance to buckling of the side skin sheets is of major importance if a maximum strength to weight ratio is obtained. The diagonal tension loads imposed by the cargo in the side skin sheets of a monocoque trailer tend to induce localized buckling loads in the sheet itself.

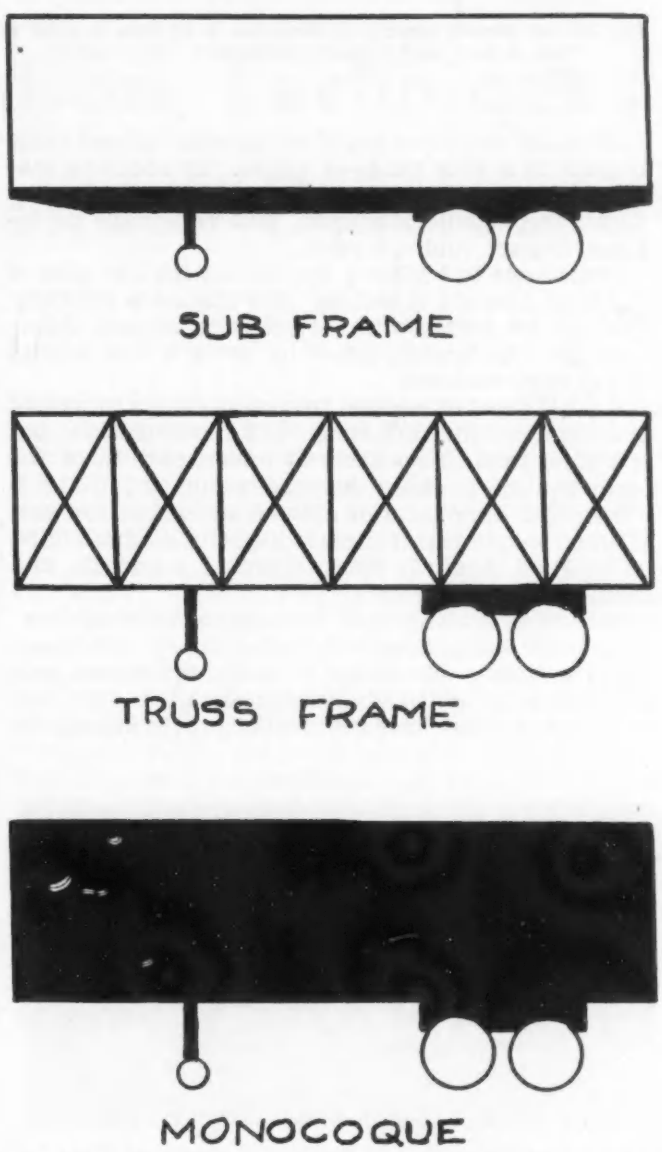


Fig. 1—Key to monocoque structural strength is fact that more area of body is stressed, as indicated by black

*Paper "Trends in Design of Lightweight Highway Equipment" was presented at SAE British Columbia Group, Sept. 9, 1948. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

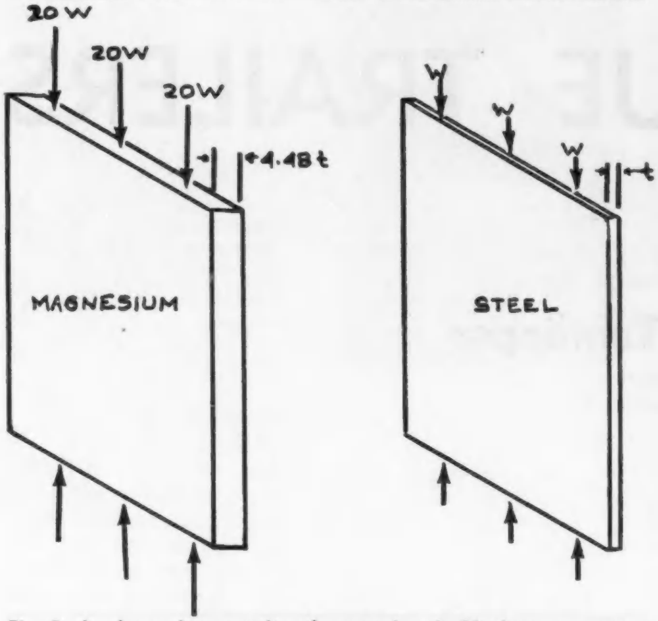


Fig. 2—Load carrying capacity of magnesium is 20 times as great as that of steel panel of equal length, width, and weight

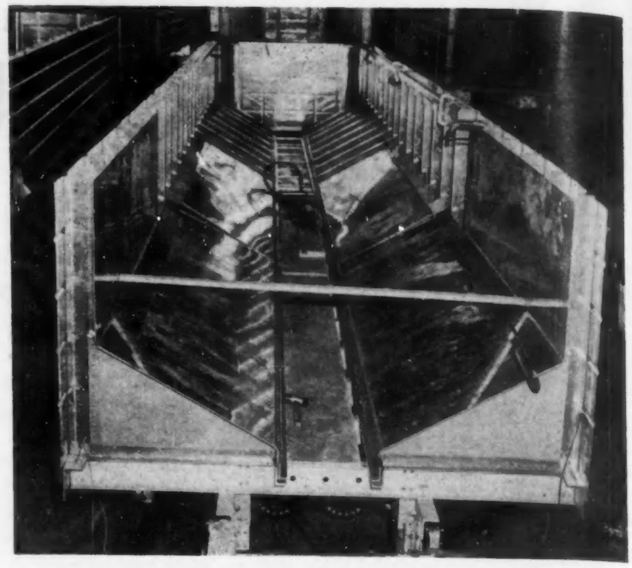


Fig. 4—Partially completed 42-ft monocoque V bottom coal trailer. Floor sills and ribs are seen forward. Conveyor belt to be installed in bottom will move at 200 fpm on roller bearing support rolls at 2-ft intervals and will unload 50,000 lb of coal in about 15 min

Suitable materials are steel, aluminum and magnesium in a wide range of alloys. In choosing material the properties to be considered are density, shear and tensile strengths, and resistance to fatigue, impact, and corrosion.

Resistance to buckling increases with the cube of the thickness of the section. Resistance to buckling, through an increase in the effective section thickness, may be accomplished by using a low density heavy gage material.

A steel sheet of a given thickness equals in weight that of a magnesium sheet 4.48 times thicker, but the steel fails in buckling at one-twentieth of the load required to cause the magnesium to fail Fig. 2.

Whether aluminum or magnesium alloys are best for monocoque van trailers is a question which must be weighed carefully from technical, economic, and maintenance viewpoints.

Ease of extruding light metals into intricate sections and with inexpensive dies is an important manufacturing advantage in using aluminum and magnesium alloys in highway trailers.

Introduction of "cast in" steel inserts in aluminum alloy castings permits a widened use of light metal castings.

Concentrated axle or king pin loads must not be imposed upon a monocoque structure unless they are distributed across a wide area of the trailer sidewall. Proper longerons should be used to distribute such loads widely into the skin sheets to prevent fatigue failure.

Monocoque construction sacrifices a great deal of its inherent effectiveness when all or a part of the roof is eliminated, as in open top trailers. Torsional rigidity is reduced to approximately one-twentieth of the standard trailer, and additional strength must be incorporated into the sidewalls to partly overcome this loss in rigidity, and to resist bulging loads.

Quality control is of more than average importance in monocoque design because of the exacting requirements of the stressed skin. Each square inch of external surface is carrying a load, and each square inch must be properly assembled and inspected.

Standard monocoque semi-trailers have been lengthened from 35 to 45 without any indication of failure after more than a year's continuous service. They have hauled payloads of 80,000 lb at high speeds Fig. 3.

Total empty weight, including extra heavy floors for general freight hauling, with 17,000 lb axles and $16\frac{1}{2} \times 7$ in. brakes is only 10,800 lb. Corrugated aluminum floors recently developed would reduce the weight another 1000 lb. This flooring, Coralite, weighs only 1.855 lb per sq ft, and in many ways surpass wood floors weighing $4\frac{1}{2}$ lb per sq ft.

Monocoque coal trailers, with V hopper bottoms, weighing 9229 lb, and 42 ft long, haul 50,000 lb of payload at high speeds Fig. 4.

Much is being done on prototype models in experimental use forecast even greater reductions in weight, better roadability, increased tire life, and almost complete elimination of troublesome—and too often overlooked—maintenance.

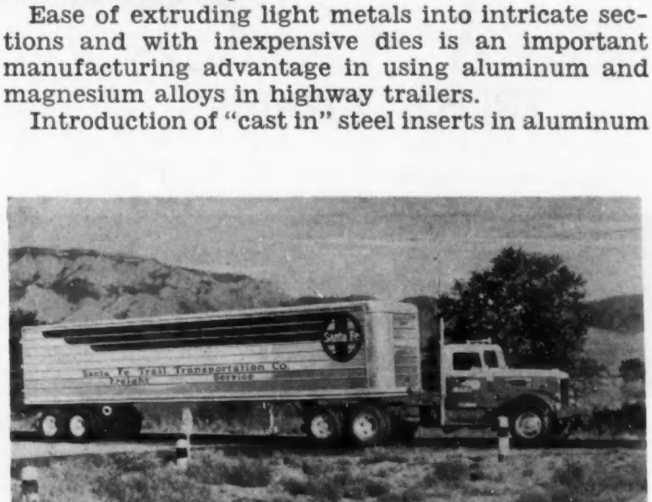


Fig. 3—Forty-five-ft monocoque trailer with two 17,000 lb axles, empty weight 10,800 lb, hauling 80,000 lb payload

● On the General Committee for the meeting were: (left to right) B. E. House, chairman, Truck & Bus Meetings Committee; M. E. Nuttila, chairman, Transportation & Maintenance Meetings Committee; E. P. Lamb, SAE vice-president for Truck & Bus; Norman Hoertz, chairman, Cleveland Section; and J. L. S. Snead, Jr., SAE vice-president for Transportation & Maintenance

NEW Transportation Techniques and Design Trends Disclosed

FULLY documented technical news about recently developed engineering products and techniques, and up-to-date trends in design of transportation equipment and components were highlighted at the SAE National Transportation Meeting, March 28 to 30, in Cleveland. More than 400 operating and design engineers met to make these new appraisals of engineering progress.

The meeting, sponsored by the Truck & Bus and by the Transportation & Maintenance Activities, reached into the third dimension of transportation with a dinner address on the Berlin Airlift by Rear-Admiral John P. Whitney. He explained the scheduling, precision instrument flying, and maintenance of history's most dramatic feat of freighting.

Toastmaster of the dinner was Donald C. Hyde, general manager, Cleveland Transit System. Cleveland Section Chairman Norman Hoertz welcomed the dinner guests and SAE President Stanwood W. Sparrow spoke briefly, stressing importance of SAE meetings and technical committee work in advancing highway transportation.

"This is a task, not for one engineer, but for many working together" he said. "The SAE has a reputation for providing a congenial atmosphere for folks who desire to work together."

Precision planning and execution of Operations Vittles has made possible shipment of 1,200,000 tons of coal, food, medical, and other supplies into beleaguered Berlin in 269 days, according to Admiral Whitney, vice-commander of the Military Air Transport Service. Average for the first three weeks in March was 5900 tons daily, about 300 more than the February daily average.

Traffic Control "Amazing"

"Amazing development of air traffic control has made this record possible," the Admiral said.

"Aircraft operate as closely as three minutes apart and with altitude variation of 500 ft between each plane. This eliminates need for 'stacking' and permits even flow of the cargo carriers to the airports."

So precise is the instrument flying that effect of bad weather on tonnage is minor, he reported.

Airplanes are used 8 hr a day, on the average. After 200 hr the airplanes get a check in Germany or England. After four of these 200-hour tours, the airplane is flown to Westover Air Force Base in Massachusetts, and goes to one of three reconditioning plants.

It then picks up a load of parts, engines, and replacement crews and is flown back to Germany. About 75 aircraft are kept flowing to ensure a full operation fleet of 225 airplanes in Germany.

At the Rhein-Main, Wiesbaden, and Berlin airports 3000 displaced persons and Germans work as stevedores under 55 officers and 295 enlisted men of the Army Transportation Corps. They have de-

veloped a split-second procedure to load and unload aircraft, at about 10 tons a load, as quickly as possible.

Daily average time to unload 10 tons from a C-45 is 12 min, the record being eight.

New Technical Developments

Technical developments revealed at the meeting centered partly around a fluid of finely powdered iron dispersed in a liquid and which is susceptible to an infinitely variable change in viscosity when subjected to a magnetic field; partly around the world's largest field dynamometer; and partly around a review of modern engine testing equipment.

The magnetic fluid, it was disclosed, was developed at the National Bureau of Standards where experimental applications have been used in clutches. But its properties, as demonstrated at the meeting, seemed equally applicable to shock absorbers which could be controlled from the dashboard, to brakes, and several other automotive components.

Background of research reported pointed to need for more work on the fluid and its ferrous content and for solving the problem of adequate sealing. None of the apparent problems, however, were thought to be insurmountable.

Wear in a clutch operating by means of this new fluid, it was said, would be imperceptible—and first costs for many projected applications should be less than that of existing equipment the magnetically operated units would replace. Either a-c or d-c current, in small amounts, can be used to excite the field.

Clutches demonstrated at the meeting operate on the principle that when the space between two parallel magnetic surfaces is filled with finely powdered iron, and a magnetic field is established between the two plates, the magnetized particles bind the plates together.

These differ from conventional magnetic clutches because they do not follow a square law wherein the torque is proportional to the square of the current. They exhibit the square-law effect to a markedly lesser degree, and in some designs the torque current curve is almost a straight line below saturation.

Static and kinetic frictions were shown to be almost identical, and breakaway torque approximates static torque. Smoothness of the new magnetic clutch operation was attributed to this fact because chattering in an ordinary dry friction clutch is largely due to the difference between static and kinetic friction.

A second new development described at the meeting was the world's largest field dynamometer. Quantities tested by this 60-ton, 1000 hp electric drive vehicle are drawbar pull, acceleration, road, track, and engine speed, and fuel flow.

A camera with synchronized flash and a moving picture camera are mounted focused on the instrument dials to record readings.

Drawbar pull is measured by a calibrated steel section carrying the force. Resistance wire strain gages are mounted on this member. Electrical output of the gages is amplified to operate the indicator.

Two strain tubes are mounted in series. The lower has a capacity up to 25,000 lb, and the high range tube measures up to 150,000 lb. The electrical hookup is sensitive enough to indicate as little as 0.000062 volts.

Acceleration of the dynamometer can be held to approximately 10% of the most sensitive scale, or 0.02 ft sec squared. Thus was disclosed a test tool

SAE President Sparrow Says:

Someone once said that, if he could write a nation's songs, he cared not who wrote its laws. Since this remark is frequently quoted it is probable that it makes some sense though I am inclined to believe that songs more frequently reflect than create the desires of a people.

Many songs reflect a yearning to be somewhere else. The bum on Broadway begs to be carried back to "Ole Virginny," and when winds blow cold all join spiritedly in swelling the chorus of "California, Here I Come."

Nor are people content with things that are close at hand. They demand products from the far corners of the globe—although it has never been clear to me just how a globe manages to get corners.

This of course all adds up to a universal craving for transportation, which gets people from where they are to where they wish to be, and gets products from where they are found to where they are used.

So far as we know, no need for the SAE was felt by those who were charged with the problem of maintaining the camels which, in ancient days, transported pilgrims to Mecca. Need for the Society came when the internal combustion engine began to play its important role in transportation.

Meetings such as this are evidence of the extent to which the Society meets that need. Problems facing those concerned with transportation and maintenance have increased rapidly in recent years.

It was not so long ago that billboards throughout the land proclaimed a person's willingness to walk a mile for a Camel. Today hardly anyone is willing to walk a block for a bus.

Today John Public wants his transportation to come to the door, to be on time, to travel fast, to travel far and to travel safely. Since the customer is alleged to be always right he is likely to get his wish.

But the task of meeting such desires is not easy. It is a task, not for one engineer, but for many engineers working together. The SAE has a reputation for providing a congenial atmosphere for folks who desire to work together.

The Society is pleased that so many of you have taken advantage of that atmosphere in working together for that industry of whose rapid progress you have reason to be proud. (Remarks by SAE President Stanwood W. Sparrow at the dinner of the SAE Transportation Meeting in Cleveland, March 29, 1949.)

● (Left to right): SAE President Stanwood W. Sparrow; Rear-Admiral John P. Whitney, vice-commander, Military Air Transport Service; Donald C. Hyde, toastmaster; and Robert Cass, general chairman of the meeting

● (Left to right): C. H. Scragg, Cleveland Section vice-chairman for Truck & Bus; R. F. Steeneck, Cleveland Section vice-chairman; and O. W. Smith, Cleveland Section vice-chairman for Transportation & Maintenance

of great interest to research and development engineers throughout the industry.

Completing the group of new technical developments outlined at the meeting was a review of modern engine testing equipment and techniques.

Importance of "know-how" was stressed. Best of instruments and equipment, it was agreed, must not be substituted for proper training of engine design fundamentals or for the exercise of good judgment. Instruments were held to be merely yardsticks to fixing the wrong thing, or fixing the right thing at the wrong time.

For example, experience alone can determine the best air-fuel ratio of a given engine. When this fact is known exhaust gas analyzers can be calibrated to show the point of the greatest fuel economy of the given engine with those predetermined characteristics.

Helpful in testing for compression leakage, it was disclosed, is a new tester consisting essentially of two gages, one calibrated in terms of cylinder bore size and the other in terms of an arbitrary rating of the cylinder ranging in ten steps. Readings up to three are considered good, from three to six fair, and the higher range poor.

With 75 psi in the air line, air passes through the first gage where a valve adjusts pressures to various bore diameters. It then enters the second gage connected through calibrated orifices. The rate at which air passes into the instrument is the reading for leakage.

Versatility of the new development is indicated by the fact that tests of leak through the engine gasket to the cooling system, leaky gaskets between cylinders, leaky inlet or exhaust valves, air leaks at breather pipe or oil filler pipe, cracked cylinders, and valve timing can be also made.

Rpm gages have been improved by electrical actuation, and a new supersensitive gage assures greater accuracy for setting governors, engine timing, idling on fluid drive cars, generator cutouts, and rpm changes from any given throttle setting.

Operating engineers were warned not to stray from manufacturers' specifications.

Design Trends Disclosed

Trends in spring design, trailer construction, and brakes, brought to the meeting up-to-date information calculated to reduce operating costs of motor vehicle transportation.

Five design trends were offered to the meeting to reduce spring cost per mile of vehicle operation. Engineers were told to consider:

- Generous spring dimensions for maximum responsiveness to road shocks,
- Minimum waste of effective spring material at axle seats,
- Full consideration of shackle angle to provide variable rate of spring deflection under varying loads,
- Use of the finest spring steels, properly heat treated, to achieve highest possible working stresses, and
- Shot peening of tension surfaces to increase spring life further.

Design engineers were urged to consider comfort of truck drivers in laying out suspension systems. Whereas modern automobiles have frequencies of

● Equipment in action enlivened the afternoon session on the first day of the meeting. Members examine some of the units used for demonstration by Speaker Jacob Rabinow, National Bureau of Standards

from 60 to 75 oscillations per minute to obtain present standards of rider comfort, most trucks have frequencies as high as 100 to 150 per minute. This makes extremely uncomfortable riding for the truck operators, the meeting was reminded.

A recent design trend reported is a single leaf spring, graduated either in width or thickness to approach the characteristics of a beam of uniform stress. Improvement in sidewise stability is indicated.

Another trend, it was disclosed at the meeting, was a three leaf spring design. It was learned that this would retain a moderate amount of interleaf friction which, in turn, would relieve the shock absorber of some of its work in damping the oscillations.

Other spring design highlights reported included:

● Buses with torsion and rubber torsion springs show almost perfect service as far as the spring elements are concerned, but the shock absorbers, shackles and associated linkage pose a maintenance cost which offsets the virtues of the torsion springs.

● Although air springs produce an excellent ride, current high costs of manufacture of these units have limited their use.

Highly important design trend in highway transportation, the meeting heard, is in improved insulation and refrigeration of trailers to handle the mounting volumes of frozen food sales.

Sufficient experience data have been obtained on fiberglass, fireproof cotton, kapok, cork, and several types of foam rubber and foam synthetics to evaluate their respective advantages.

Operating engineers were warned of the importance of vermin, bacteria, and water resistance of insulating materials, whether ice, dry ice, or an ice making machine is used as the primary refrigerant.

Sandwich construction, where two light sheets of aluminum or steel are bonded on each side of a 2 in. flat of foam material, was disclosed as an important trend. Adopted from aircraft construction, such designs offer greater strength at lighter weight than the more conventional insulation of a conventional truck.

Fundamental requirement, the meeting was told, is to specify in detail conditions under which proposed refrigerated equipment is to work, including facilities at terminals, temperature bands at which perishables must be transported, length of routes and ambient temperature ranges of the routes.

It was pointed out that physical characteristics of insulating materials must be carefully weighed to achieve best results. The meeting was told that foamed styrene, for example, weighs about 2 lb per cu ft, has a K factor of 0.3, and a loading capacity of 20 psi. A cast cellular rubber brick with a K factor of 0.22, weighs 5 lb per cu ft, and has a loading capacity of about 40 psi.

Unmistakable is the trend to light metal alloys in trailer design. Much of this advance, the meeting disclosed, has resulted from improved techniques in fabricating light metals, and a better understanding on the part of operators that often equipment built of aluminum and magnesium sheet, extrusions, and castings require more exacting maintenance than trailers built largely of steel.

Relative ease with which light metal alloys can be extruded into intricate shapes gives them cost advantages over steel because judicious use of a wide range of stock shapes eases fabrication.

Where hauls are long and weights of payload are up to legal maxima, savings in weight by use of light metals permit additional loading which have written off the higher initial cost within seven to eight months, engineers learned.

Brake Discussion Lively

The subject of trends in heavy duty brakes unloosed the most vigorous discussion of the three-day transportation meeting, and was the final contribution to the disclosure of design trends in highway transportation.

Although sharp differences of opinion mark most SAE meetings on the subject of brakes, agreement was reached on these salient points:

● Engine horsepowers are going up, not down. Hence the brake heat dissipation problem is increasingly difficult to solve,

● More light should be thrown upon the questions of friction coefficients and how they vary with temperature changes, the subject of weight transfer, and unequal braking, and

● Water and air cooling, use of aluminum drums and aluminum bonded fins, and investigation of basic design changes offer promising roads to improved braking systems.

Coupled with the trend to larger engines is greater vehicle speeds which compound the brake heat-dissipation troubles. Some experience was reported with the vagaries of friction coefficients, and a part of the audience doubted the validity of a single factor for varying conditions of braking surface speeds, temperatures, and pressures. This led to speculation about outside aids to cooling such as water and air, and mechanical methods of self aid, such as fins, baffles, and inducing slipstream cooling or fans to keep the drums and components reasonably cool during long, hard applications of brakes on heavily laden vehicles.

The meeting heard a review of current development and projected glimpses into tomorrow, design engineers and operators swung into detailed discussion, and the technical sessions closed on the note of continued inquiry into one of the major design problems facing the automotive engineer today.

Comparisons of Current Helicopter Configurations

Based on paper by

R. H. PREWITT

Prewitt Aircraft Co.

WITH the ever increasing speeds and long range of air transport services, the unique performance of helicopters will play an important part in short haul transport.

● Present air mail operations of helicopters will gradually develop into carrying passengers.

● With increasing distances between airports and centers of business districts of cities passenger-ferrying helicopters will be required. New York and Pittsburgh have completed such projects.

● Engineering improvements are being carried on at a rapid rate.

Survey of 73 helicopter models, of which 52 are American and 21 foreign, several of which are behind the "iron curtain," discloses an interesting variety of configurations, Figs. 1 to 6. Jet powered rotors are in an early stage of development.

Powerplant arrangements include conventional aircraft engines modified to meet power transmission and cooling requirements.

In some instances successful engine cooling has been attained by substituting exhaust jets for an engine driven fan. This diverts fan power to useful lift.

Ram jet powerplants mounted at tips of blades, and on or off burner jets at the tips of blades supplied with compressed air from a fuselage mounted compressor, constitute the main efforts for jet propelled helicopters.

Today's scheduled helicopter operations parallel taxicab charges of 25¢ per passenger mile. These costs are heading downward. Some manufacturers of the larger machines calculate operating costs, less overhead, at 5¢, or 12¢ per passenger mile including overhead.

A 6¢ passenger mile rate would put a pack of helicopters into bus and feederline operations.

A suitable helicopter engine would provide for higher rotational speed of

Fig. 1—Side-by-side rotors may prove superior for high rate of climb. It has exceptionally high parasite drag. Support and driving mechanism for separate rotors increases structural weight

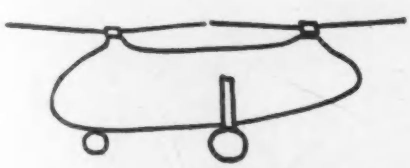


Fig. 2—Fore and aft rotor design provides larger fuselage for passengers and cargo. Frontal area drag is less than in Fig. 1, but its rate of climb might be expected to be below normal

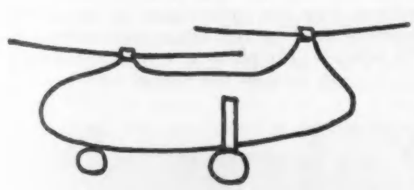


Fig. 3—Fore-and-aft rotor helicopter with aft rotor above level of foreward rotor has increased effective aspect ratio because aft rotor is appreciably higher when the machine tilts forward for forward flight

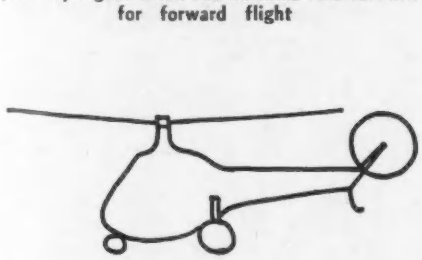


Fig. 4—Single main rotor with tail rotor is simpler, and total weight and cost can be reduced. More effective for smaller helicopters. For constant disc loading size of tail rotor should be increased more rapidly than that of main rotor due to the latter's slower turning speed

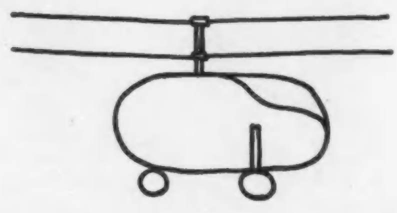


Fig. 5—Coaxial helicopter with rotors turning in opposite directions has advantage of centrally located power transmission and control systems. Increased efficiency is due to effective vertical displacement of rotors relative to each other. Lower rotor recovers race rotation loss of upper blades

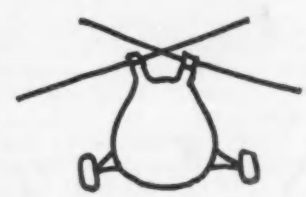
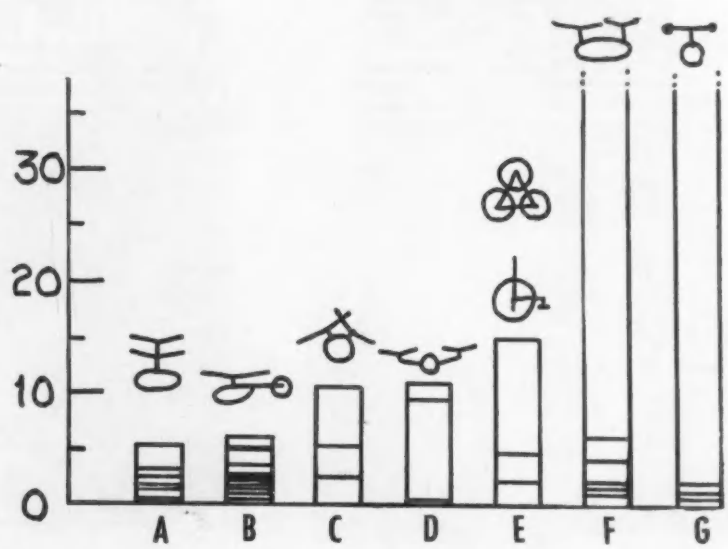


Fig. 6—Synchropter has rotor mounted angularly. Here rotor hubs must be placed farther from the ground for adequate ground clearance of blade tips. In this, as in Fig. 5, parasite resistance is increased, as is center of gravity height

Fig. 7—Cross weight in 1000 lb of various current helicopter designs. Column A shows counter rotating coaxial; B, (Fig. 4); C, synchropter, Fig. 6; D, lateral rotor type, Fig. 1; upper diagram of column E represents the British 3-rotor Whitehorse and lower the British Gyrodyne, holder of speed record, which assumes about one half the characteristics of helicopter and half autogyro in flight; F represents a fore and aft rotor design, and G jet-propelled rotors



the blade at higher altitude. To date there is no such engine.

Each of the many helicopter configurations have certain natural advantages and disadvantages.

Jet propelled rotors have the advantage that no special torque-correcting device is needed for a single rotor machine. It can be built lightest of all because it needs no engine or power transmission system. Its disadvantage is extremely low fuel efficiency.

Gross weights of some of today's helicopter designs are compared in Fig. 7, and advantages gained when several manufacturers order given components from a single vendor are shown in Fig. 8.

A first cousin of the helicopter is the convertaplane. It includes all the variations between the operational characteristics of the fixed wing airplane and those of the helicopter or autogyro. Much work has been done on this design by Gerald Herrick and others, but much research and development must be done to arrive at a practical useful machine. (Paper "Today's Picture in Helicopters" was presented at SAE Annual Meeting, Detroit, Jan. 13, 1949. This paper is available in full in mimeographed form from the SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

New Heavy-Duty Lube Extends Diesel Life

Based on paper by

J. A. Edgar,
J. M. Plantfeber,
and R. F. Bergstrom
Shell Oil Co., Inc.

(This paper will be printed in full in SAE Quarterly Transactions)

A new special heavy-duty lubricant for diesels has several times the detergency, oxidation stability, additive stability to high piston temperature, and antiwear property of conventional oils of this type.

Field experience has shown that under certain operating conditions normal heavy-duty oils do not come up to expectations. The special compounded lube has corrected corrosive wear and fouling—whether these conditions stemmed from unusual operating conditions, use of sulfurous fuels, or both.

A case in point is heavily-loaded hot-

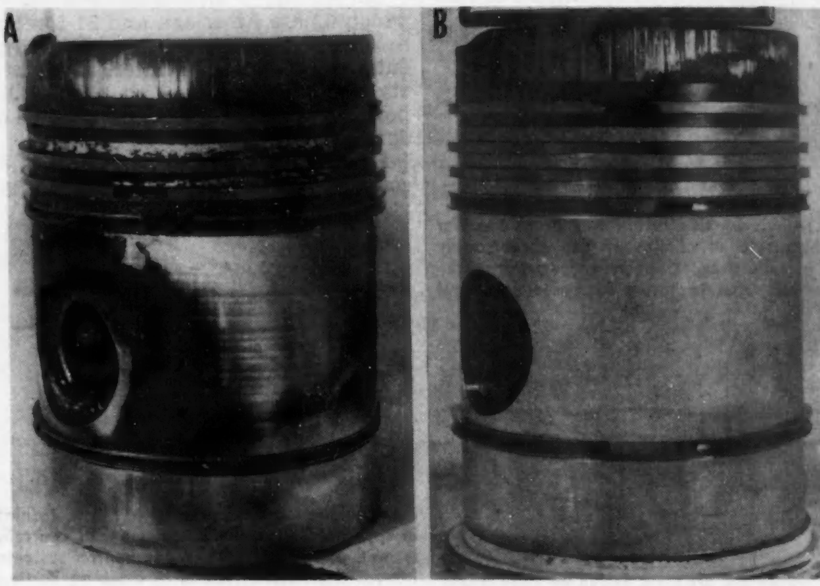
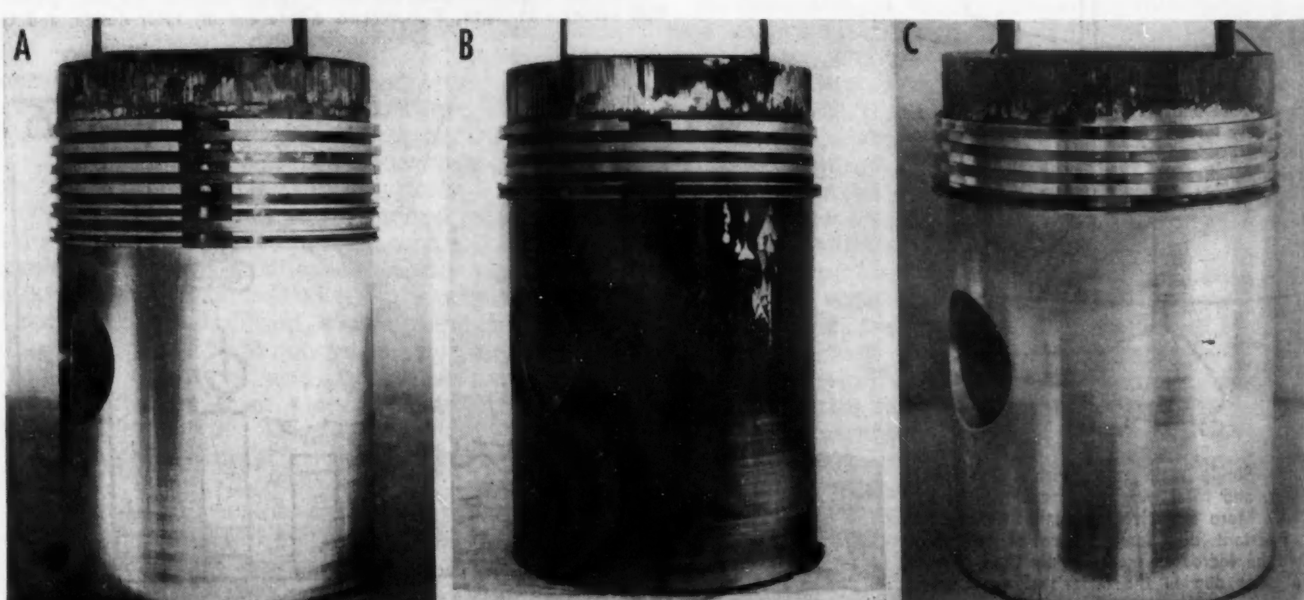


Fig. 1 (above)—The new special heavy-duty oil improves cleanliness and wear of diesels, these pistons from a pumping engine show. The one at left ran 470 hr on conventional oil and 1094 hr on the new oil. But note the deposit remaining from the conventional oil, despite the cleaning resulting from subsequent operation on the new oil. The piston at right operated twice as long on the special lube.

Fig. 2 (below)—The new oil also combats effects of sulfur in fuel. The piston in "B" ran on a 0.5% sulfur fuel and conventional heavy-duty oil. Running the same engine on the same fuel, but this time with the new oil, yielded the clean piston in "C" and about one-tenth the cylinder wear with the conventional oil. Wear was less than in the engine from which the piston in "A" was taken, which operated with a premium grade fuel, containing 0.2% sulfur, and on normal oil.



running engines which had excessive engine deposits and unexpectedly high wear rates, such as the one in **Fig. 1.** The pistons shown are from a pumping engine, which ran under a fairly heavy load, with cooling water outlet temperature of about 200 F. Fuel used was typical of Los Angeles Basin production.

When lubricated with conventional heavy-duty oil, this engine required overhauling at about 1500-hr intervals. Ring sticking and piston fouling normally occurred during this operating period and high cylinder wear—averaging 0.017 in. per 1000 hr—was observed.

With the special heavy-duty oil, as the piston at right shows, piston fouling was eliminated and wear reduced to 0.0005 per 1000 hr, a ratio of 34 to 1. This piston was operated 3172 hr.

The piston at left in Fig. 1 gives some indication of deposits previously obtained with normal oil. There still remain heavy deposits after 470 hr of operation with conventional heavy-duty oil, even after evidence of some cleanup from a subsequent 1094 hr of operation with special heavy-duty oil.

Other results obtained with engines in pumping service are shown in **Fig. 2.** Excellent performance in "A" came from a moderately loaded engine, using good heavy-duty oil and a premium grade fuel containing 0.2% sulfur. Cylinder wear rate was 0.0012 in. per 1000 hr—a creditable figure.

But a neighboring engine, operating at rated load and the same temperature, and using a commercial fuel with 0.5% sulfur, had serious difficulty, as shown in "B" of Fig. 2.

A direct comparison then was made in the second engine by substituting the special lubricant, leaving other conditions unchanged. "C" depicts the high level of cleanliness achieved. Cylinder wear rate is 0.0004 in. per 1000 hr —one-tenth that with the conventional oil in the same engine and lower than the result with normal oil and a premium fuel. This exemplifies the practical elimination of corrosive cylinder wear.

Diesels used in highway hauling also can benefit from the new special oil, as shown in **Fig. 3** by the ring belt areas of two pistons from an engine in such service. The piston at left operated 100,000 hr on conventional heavy-duty oil. One ring stuck and the rings wore so that no tool marks remained. Operating 80,000 miles on the special oil, the piston in the photograph at right shows ring tool marks on all the compression rings and no rings stuck.

Allowing for a mileage advantage and a new piston design with the special heavy-duty oil, there is still very low piston-ring wear and a high degree of cleanliness.

Fig. 4 illustrates correlation of general engine cleanliness with piston condition. Valve guide sleeves with pistons from their respective engines show the same relative condition. Sleeves in top photograph were so heavily lacquered that they could not rotate, while those on the bottom are free of varnish. (Paper "Special Heavy-Duty Type Engine Oils," was presented at SAE National Fuels and Lubricants Meeting, Tulsa, Nov. 4, 1948. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

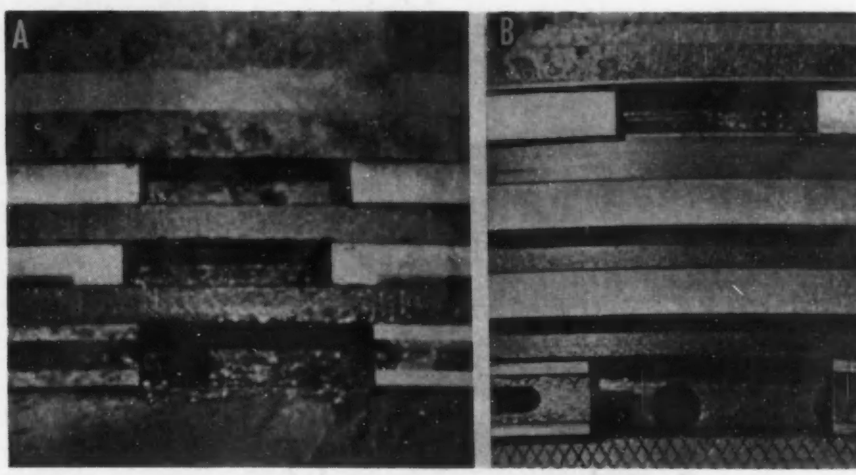
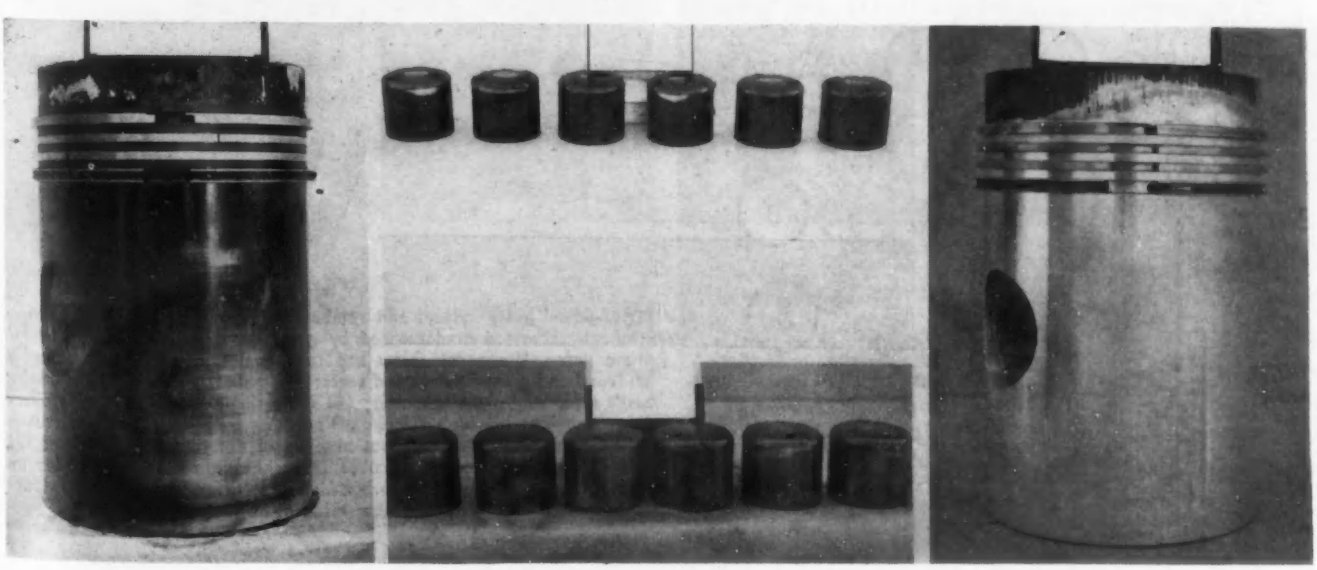


Fig. 3 (above)—Automotive diesels operating on high sulfur fuels also benefit from the special heavy-duty lube. The piston ring-belt area in "A" comes from an engine using conventional oil; that in "B" from an engine operating with the new oil.

Fig. 4 (below)—Note the deposits on the piston at left and top set of valve sleeve guides, all from the same engine, which operated for 2829 hr with 0.5% sulfur fuel and conventional heavy-duty oil. Average cylinder wear rate was 0.004 in. per 1000 hr. The piston at right and the bottom set of valve sleeve guides are from an engine in equivalent service, but operating 1260 hr on a 0.6% sulfur fuel and the special lubricating oil. Average cylinder wear rate here was 0.0005 in. per 1000 hr. Note the freedom from deposits and ring sticking in this case.



Simplicity Is Basis Of Sheppard Diesel Design

Based on paper by

R. H. SHEPPARD

R. H. Sheppard Co., Inc.

FIRST and foremost feature of the Sheppard diesel engine is simplicity of its design.

During experimental development many useful and good designs were discarded because they were believed to be too complicated and intricate for widespread use by untrained personnel.

The production diesel powerplant is simple enough to be readily understood and serviced by the average maintenance man under field conditions.

This philosophy naturally leads to the engine's next most important feature, which is ruggedness of construction.

Freedom from a multiplicity of small delicate parts has been achieved. Where a part must be small, it is kept as large and rugged as is compatible with good performance and good engineering practice.

The fuel pump in the injection system consists of individual variable stroke plungers varied to meet the governor requirements. This is accomplished by a revolving series of moveable cams, one for each cylinder. These are retracted or protruded by a rotating inclined plane positioned by the governor.

The fuel pump camshaft is a composite shaft consisting of an inner and an outer member. The inner shaft contains the inclined planes and moves axially. It is actuated through the roller plate and governor rollers by the governor, which is mounted on the outer shaft.

As the shaft rotates the cams strike the fuel pump cups and, because the lower end of the plunger rests on the bottom of the cup, injection takes place as the cup rises. Constant velocity motion is approximated during the effective part of the stroke because the nose of both the cam and cup are contour ground.

Vertical adjustment of the barrel increases or decreases the amount of fuel injected because the inlet port is in the side of the fuel pump. Hence the pump is balanced without disassembly.

Secret of long life of the check valve in the injector is the lightly loaded seat. It is not unusual for these valves to give up to 20,000 hr of service without attention of any kind.

The governor achieves great sensitivity largely due to the fact it does not have to overcome the friction of rest. As a result, speed adjustments as small as one or two revolutions per minute can be made.

Another feature of the fuel pump and governor system is that when running

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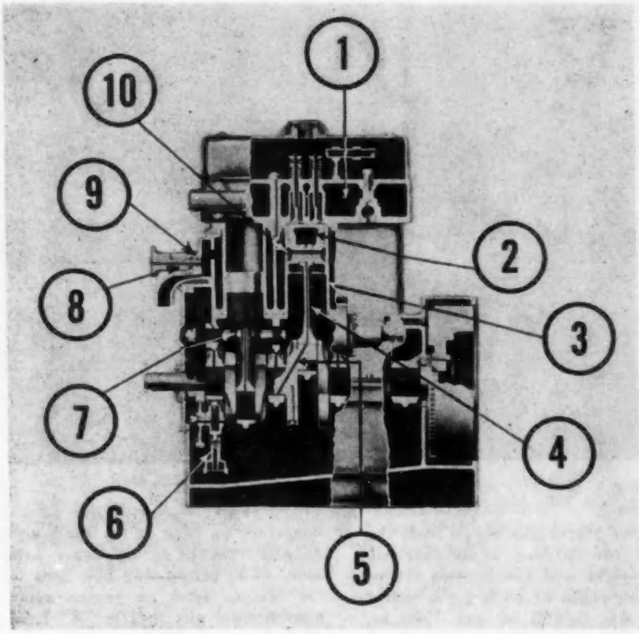


Fig. 1—Cross section of the Sheppard diesel. (1) Generous cooling areas in head reduce valve temperatures; (2) pistons are oil cooled. Directionally-formed spray is distributed in relation to pattern of heat beneath combustion chamber; (3) removable wet type cylinder liners; (4) connecting rods matched in sets both for weight and length; (5) sturdy crankshaft counterbalanced for smooth operation, and rifle drilled for full pressure lubrication; (6) lube oil pump insures generous supply to all engine parts; (7) camshaft precision ground on wearing surfaces and hardened; (8) fan has sealed self-lubricating bearing; (9) pump seal seldom requires adjustment, and (10) full-floating wristpins distribute bearing wear, increase bearing life

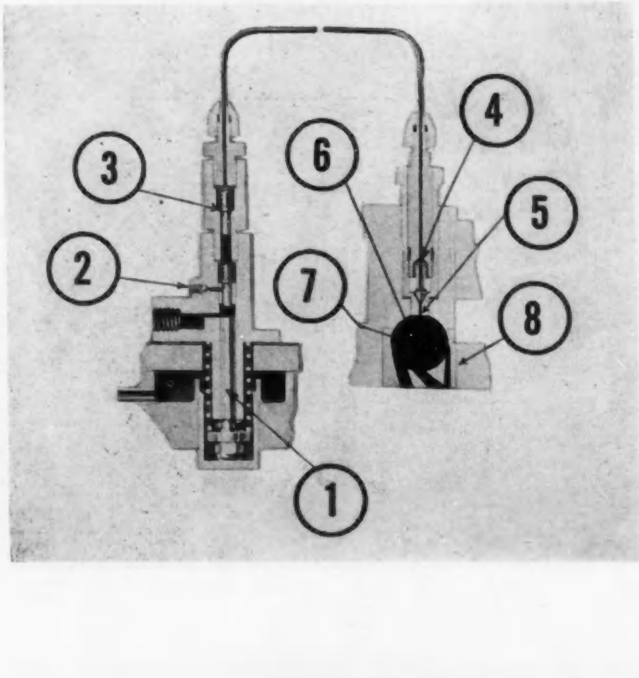


Fig. 2—Fuel pump system and combustion chamber. (1) Pump plunger stroke is increased or decreased by governor according to load; (2) to prime pump, loosen set screw and turn engine over a few times; (3) hardened and ground valves and seats; (4) complete check valve assembly easily replaceable as single unit; (5) large single orifice of nozzle eliminates clogging; (6) high-velocity air entering from three passages creates optimum turbulence and assures absence of detonation and smoke; (7) fuel entering combustion chamber is broken into minute particles for proper combustion, keeping carbon deposits to minimum, and (8) lower two-thirds of chamber made of heat resisting steel

TECHNICAL COMMITTEE PROGRESS

Handbook To Tell About Tool Steels

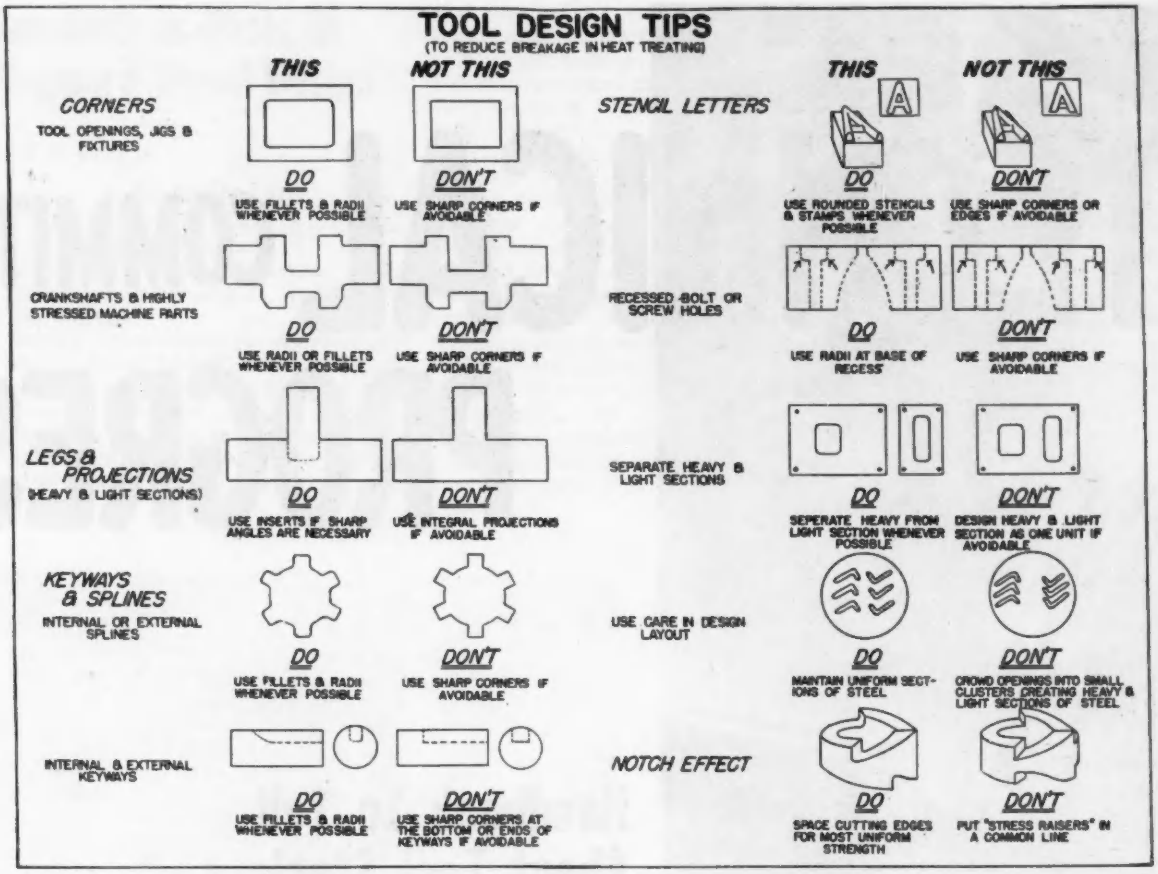
A RECENTLY-APPROVED report on tool and die steels will be published in the 1949 SAE Handbook. Covering selection and heat-treatment of tool steels, this report was prepared by the Tool and Die Steels Division of the SAE Iron & Steel Technical Committee.

Selection of tool steels, says the report, depends on the type of application—such as cutting, shearing, forming, drawing, extrusion, rolling, and battering. Cutting tools include drills, taps, broaches and hobs. Blanking and trimming dies and punches are typical of shearing tools. In the forging tools group are forging dies, cold-heading dies, and die-casting dies. Included in battering tools are chisels and all forms of tools involving heavy shock.

The report labels as important the metallurgical properties of each of these groups. It points out, for example, that cutting tools must have high hardness, high resistance to softening effect of heat, and high wear resistance. Shearing tools require high wear resistance combined with fair toughness. These characteristics must be properly balanced—depending on tool design—with thickness of stock being sheared and temperature of the shearing operation.

High toughness and strength are required by forming tools and many must have maximum resistance to heat softening. High toughness is a must with battering tools.

In the light of these requirements, the report considers these as prime selective factors for tool steel use: hardness, strength, toughness, wear re-



sistance, and resistance to heat softening. Other properties also demand serious consideration. Included are permissible distortion in hardening, permissible surface decarburization, hardenability, machining and grindability.

To help evaluate these tool steel properties, the 1949 SAE Handbook will rate the various tool steels on the basis of poor, fair, good, and best. The steels are broken down into these classifications: water-hardening, oil-hardening, air-hardening, shock-resisting, hot-work, and high-speed tool steels.

Another part of this report discusses the relation of design to heat-treatment of tools. This part aims at developing an understanding between designer and heat treater to avoid distorting or cracking during heat-treating due to faulty design.

Successful design from this standpoint, says the report, can be boiled down to this basic principle: A part is properly designed from the standpoint of heat treatment when the entire piece may be heated and cooled at about the same rate during the heat-treating operation.

It is pointed out that this kind of perfection is unattainable since even a sphere's surface cools more rapidly than its interior. But the designer should try to shape his parts so that they will heat and cool as uniformly as possible.

The greater the temperature difference between any two points on a given part during quenching and the closer these two points are together, the greater will be internal strain and the poorer the design.

Design Errors Noted

It is conceded that some shapes are almost impossible to harden because of abrupt changes in section. But a certain latitude in design is recognized when using oil-hardening or air-hardening steel. Errors in design, the report continues, do more than affect internal strains during hardening. A sharp angle greatly concentrates service stresses. Design may be entirely responsible for concentrating service stresses at a point already weakened by internal strains produced during hardening.

In addition to the design "do's and don'ts" shown above, the report also gives forging, normalizing and annealing treatments for principal types of tool and die steels.

Temperature at which to start forging is given as a range—the high side for large sections and heavy or rapid reductions, the low side for smaller sections and lighter reductions. Annealing temperature also is given as a range.

Correction

In the article on "Additive Treated Crankcase Lubricants," Table I, p. 30, February, 1949, issue of the SAE Journal, stated that Caterpillar Tractor adopted copper-lead bearings in 1937.

This should have read:

"Caterpillar's test procedure was modified to require that heavy-duty oils be non-corrosive toward copper-lead bearings."

Prop Shaft End Standards Revised by SAE Committees

Heading up the groups that developed the newly-issued propeller shaft standards are R. P. Lambeck (left), Hamilton Standard Propellers, chairman of the SAE Propeller Standards Committee, and G. N. Cole, Pratt & Whitney Aircraft, chairman of the SAE Committee on Standard Components for Aircraft Engines

TWO up-to-date standards for dual rotation propeller-shaft ends, prepared by the SAE Propeller Standards Committee and the SAE Committee on Standard Components for Aircraft Engines, were issued the first of this month. They are: (1) Aeronautical Information Report AIR No. 26, Nos. 70-90, 70L-90, 80-100, and 80L-100 Propeller Shaft Ends, Dual Rotation (Propeller Supplied Bearing), and (2) Aeronautical Recommended Practice 375A, Propeller Shaft Ends, Dual Rotation (Propeller Supplied Bearings).

ARP 375, initially issued Nov. 1, 1948, included dimensions and data for Nos. 70-90, 70L-90, 80-100, and 80L-100 dual rotation shafts. The 70-90 and 70L-90 shafts were intended for use with engines of 8000 and 12,000 hp. However, it became apparent that the 70 and 70L shafts in these combinations were critically loaded due to gyroscopic and 1XP continuous bending moments, encountered in modern high-speed, high wing-loading aircraft.

Therefore, dimensions for the 80-90 shafts were developed for inclusion in the revised ARP 375. Since it appeared that the 70-90, 70L-90, 80-100, and 80L-100 shafts would not be used, it was decided to delete them from ARP 375 and to include them in AIR No. 26.

Standards for dual rotation propeller shaft ends were undertaken by the two SAE Committees two years ago at the request of the Aircraft Industries Association. Although AIA asked for development of standards for the 80-100 shaft size combinations only, the conferees found it desirable to include also the 70-90 combination.

Big demand for these standards brought about their initial release late last year, despite the rapid advance of the art. The newly-issued versions are in line with current industry practice.

In addition to the shaft ends standards, the Committees are developing one for the engine thrust bearing nut. Aim here is to standardize only that part of the nut at which the wrench is applied. Significant benefit seen for this work is standardization of tools, which would be a boon particularly to the military services.

Helicopter Reports

THE SAE Helicopter Committee recently completed two reports aimed at aiding the designer. They are:

1. Aeronautical Information Report AIR No. 19—A Criteria For the Fatigue Testing of Rotor Blades. This report is intended as a guide—based on current knowledge and experience—for use in fatigue testing of helicopter rotor blades. The Committee considers this a progress report, in the light of rapid obsolescence of rotor blade construction methods and testing procedures, subject to early revision.
2. Aeronautical Information Report AIR No. 21A—The Definition of Design Load Conditions for the Helicopter Rotor. Stemming from a continuing research program of the Committee, this report lists all essential load conditions for analysis of a helicopter rotor. It defines each condition numerically and provides a basis for structural analysis of the helicopter rotor.

Technishorts . . . .

LOADING HEIGHTS: The SAE Transportation and Maintenance Technical Committee is organizing a subcommittee to study loading platform heights, maneuvering distances, and terminal dimensions. The wide variations in these dimensions at loading docks throughout the country pose a tough problem for common carriers of dry freight.

H-BAND STEELS: Members of the Hardenability Division, of the SAE Iron & Steel Technical Committee, recently discussed the possibility of achieving better hardenability control by going to a narrower carbon range with accompanying wider manganese and chromium ranges. Advantages of closer control of carbon content versus depth hardenability was said to depend on end use of the steel part. Wider chromium and manganese ranges are required by the steel-maker in adjusting steel chemistry for these two elements. Peformance in user's processing of H-steel heats, off in manganese and chromium, will be studied to evaluate effect of these variations and results reported to the Division.

HELICOPTERS: The SAE Helicopter Committee has arranged for a meeting in New York on May 11 to facilitate attendance of its members at the Annual Helicopter Forum, of the American Helicopter Society, on the following three days. This was announced by R. H. Prewitt, chairman of the SAE Helicopter Committee and a past-president of the American Helicopter Society.

MASTIC COATINGS: The SAE Passenger Car Body Engineering Committee is developing specifications for mastic sound deadeners and underbody coatings for motor vehicles. Materials being covered are divided into two groups: (1) Types, based on the decay rate (decibels per sec at 70F) and (2) Classes, based on the percentage of solids contained in the material. The SAE Non-Metallic Materials Committee is assisting in this project.

SAE Summer

June 5-10, 1949

Sunday, June 5

7:00 p.m. BANQUET

A Big Family Dinner Party

WILLIAM K. CRESON, Master of Ceremonies

Presents

THE McPHEE FAMILY

a quartet you will long remember

8:30 p.m.

ROBERT INSLEY, Chairman

"THE SPACE SHIP AND THE MAN-MADE MOON"

DR. GERALD WENDT

Editorial Director, Science Illustrated

Monday, June 6

9:30 a.m.

M. D. GJERDE, Chairman

Application of Equilibrium Air Distillation to Gasoline Volatility Performance Calculations (A CFR-CRC Report)
—E. M. BARBER, The Texas Co., and J. H. MACPHERSON, JR., Calif. Research Corp.

Effect of Oil Viscosity on Engine Performance
—J. W. LANE, Socony-Vacuum Oil Co., Inc.; D. S. CHATFIELD, Socony-Vacuum Laboratories

(Sponsored by Fuels and Lubricants Activity)

ROUND TABLES

2:00 p.m. to 4:00 p.m.

SUBJECT	LEADER
Personal Plane Engines and Accessories	C. T. DOMAN, Air-cooled Motors, Inc.
Problems with Cast-Iron Parts	F. J. WALLS, International Nickel Co., Inc.
Injury Done to Metals by Grinding	A. G. HERRESHOFF, Chrysler Corp.
Engine Lubrication: Engine Design versus Oil Compounding	Chairman: H. L. MOIR, Pure Oil Co. Leaders: W. B. BASSETT, E. F. COLLINS, H. M. GADEBUSCH, LEON GERSHBEIN, W. S. JAMES, F. F. KISHLINE, H. O. MATHEWS, J. W. PENNINGTON, A. C. PILGER, V. G. RAVIOLO

8:30 p.m.

A. M. ROTHROCK, CHAIRMAN

Compounding of Piston Engine
—F. J. WIEGAND and M. R. ROWE, Wright Aeronautical Corp.

Compound Engine Systems for Aircraft
—E. J. MANGANIELLO, L. V. HUMBLE and D. S. BOMAN, National Advisory Committee for Aeronautics

(Sponsored by Aircraft Powerplant Activity)

Tuesday, June 7

9:30 a.m.

V. M. DARSEY, Chairman

SYMPOSIUM ON CORROSION

Corrosion Problems of the Automotive Engineer
—F. L. LaQUE and E. J. HERGENROETHER, International Nickel Co., Inc.

Corrosion of Electroplated Steel in Automotive Applications
—H. A. PRAY, Battelle Memorial Institute

Service Tests Solve Aluminum Cylinder Head Corrosion Problems
—M. W. DAUGHERTY and R. F. KOENIG, Aluminum Co. of America

(Sponsored by Engineering Materials Activity)

ROUND TABLES

2:00 p.m. to 4:00 p.m.

SUBJECT	LEADER
Gas Turbine Fuels	A. L. POMEROY, Thompson Products, Inc.
How to Prevent Wear (1:30-3:00 p.m.)	W. F. JOACHIM, U. S. Naval Engineering Experiment Station
What the Engine Has to Say About Fuels and Lubricants (3:00-5:00 p.m.)	F. G. SHOEMAKER, Detroit Diesel Engine Div., General Motors Corp.
Fatigue and Service Testing of Automotive Parts	A. F. UNDERWOOD, Research Laboratories Div., General Motors Corp.
Machining and Machinability of Iron and Steel	O. W. BOSTON, University of Michigan
Mechanical versus Chemical Octane Numbers and High Compression Engines	Co-Chairmen: A. O. WILLEY, Lubri-Zol Corp., E. N. COLE, Cadillac Motor Car Div., General Motors Corp. Leaders: KENNETH BOLDT, D. F. CARIS, H. J. GIBSON, E. P. GOHN, W. M. HOLADAY, J. G. MOXEY, Jr.
Low-Pressure Tires	J. E. HALE, Firestone Tire and Rubber Co.

Meeting

French Lick Springs Hotel

French Lick, Indiana

8:30 p.m.

F. A. ROBBINS, Chairman

Free Piston Gas Generator Brightens Turbine Future
—Commodore L. F. SMALL, Lima-Hamilton Corp.

Compound Powerplants
—P. H. SCHWEITZER, Pennsylvania State College; J. K. SALISBURY, General Electric Co.
(Sponsored by Diesel Engine Activity)

Wednesday, June 8

9:30 a.m.

R. P. LEWIS, Chairman

Hydrodynamics of the Hydraulic Torque Converter
—E. W. SPANNHAKE, Consulting Engineer

Packard Automatic Transmission
—J. G. VINCENT and F. R. McFARLAND, Packard Motor Car Co.
(Sponsored by Passenger Car Activity)

■

2:00 p.m. French Lick Airport

SAE Automotive Circus

- Cavalcade of new passenger cars
- Truck "Roadeo"
- Airship demonstrations
- Model jet plane demonstrations

Follow the band and join the parade to the French Lick Airport. Transportation will be provided.

8:30 p.m.

S. W. SPARROW, Chairman

Gas Turbines in Automobiles—A Study Comparing the Calculated Performance of a Gas Turbine with Present Automotive Engines
—W. A. TURUNEN, Research Laboratories Div., General Motors Corp.
(General Session—sponsored by SAE Meetings Committee)

Thursday, June 9

9:30 a.m.

F. B. LAUTZENHISER, Chairman

An Evaluation of Present Trends Toward Larger Powerplants
—M. C. HORINE, Mack Manufacturing Co.
(Sponsored by Truck and Bus Activity)

ROUND TABLES

2:00 p.m. to 4:00 p.m.

SUBJECT	LEADER
Body Insulation and Reduction of Noise Level	L. M. BALL, Chrysler Corp.
Pros and Cons of Increased Glass Area	C. E. HEUSSNER, Chrysler Corp.
Influence of Oil Filters and Air Cleaners on Oil Consumption	W. S. JAMES, Fram Corp.
Passenger Car Brakes	V. P. MATHEWS, Moraine Products Div., General Motors Corp.
Mechanic Training	A. W. NEUMANN, Willett Co.
Lag in Power Brakes—Application and Release	JULIUS GAUSSOIN, Silver Eagle Co.
Motor Vehicle Operating Costs	E. W. TEMPLIN, Los Angeles Department of Water and Power
Preventive Maintenance Practices	H. O. MATHEWS, Standard Brands, Inc.
Fifth Wheel Location	F. B. LAUTZENHISER, International Harvester Co.

8:30 p.m.

J. W. GREIG, Chairman

Manufacturing an Automobile in Australia
—W. E. HILL, General Motors Overseas Operations
(Sponsored by Body Activity)

Friday, June 10

9:30 a.m.

J. L. S. SNEAD, Jr., Chairman

Instrumentation for Trucks and Buses
—W. H. FARR and G. E. COXON, Stewart-Warner Corp.
(Sponsored by Transportation and Maintenance Activity)

About

J. H. HUNT, director of the New Devices Section, General Motors Corp. since 1932, has retired. He has taken up new duties as a consultant to GMC Vice-President of Engineering J. M. CRAWFORD.

SAE President in 1927, Hunt joined the Dayton Engineering Laboratories Co. in 1913. There he remained until that company was absorbed by General Motors in 1918, following which time he occupied various responsible engineering and research positions in the Corporation

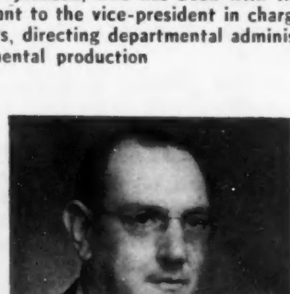
HARRY C. DUMVILLE, who has been assistant director, becomes director of General Motors Corp. New Devices Section, succeeding J. H. Hunt. Dumville, a graduate of Rensselaer Polytechnic Institute, Troy, N. Y., joined General Motor's as a junior engineer in the Research Laboratories and was transferred to the New Devices Section in 1936. In 1943 he was assigned to aircraft work on the Fisher Body central engineering staff. He returned to the New Devices Section as assistant director in 1944

BENNING

JOHNSON

WALTER F. BENNING has been appointed chief engineer of Willys-Overland Motors, and PHILIP C. JOHNSON has been appointed assistant chief engineer, DELMAR G. ROSS, vice-president in charge of engineering has announced. Benning joined Willys-Overland in 1944 as truck engineer to handle work in connection with the wartime Jeep, and has been technical assistant to the vice-president in charge of engineering for the past two years. Johnson, who has been with the company since 1937, has been assistant to the vice-president in charge of engineering for the last four years, directing departmental administration, labor relations, and experimental production

DeCAVITTE



RUMFORD

FRANK L. DeCAVITTE has been appointed factory manager of the Plymouth plant of Chrysler Corp. at Detroit. He comes to Detroit from Evansville, Ind., where he had been plant manager of the Chrysler Corp. Evansville plant. GEORGE H. RUMFORD, JR., superintendent of the Evansville plant, has been promoted to plant manager, succeeding DeCavitte

JOSEPH GESCHELIN, Detroit editor of Chilton Publications, and SAE past vice-president was interviewed on Station WJR, Detroit, on March 12 on that station's weekly industrial hour. Object of this interview was to give the public some impression of the production clinics held at the SAE Passenger Car, Body, and Production Meeting, and to indicate what they mean to the car owner.

CHRISTOPHER DYKES was recently promoted to assistant technical development director of engineering at British Overseas Airways Corp. in London. His former post was deputy chief project engineer.

ERLE J. HUBBARD has accepted a position as foundry manager, Janney-Cylinder Co. in Philadelphia. He has resigned his position as foundry superintendent for Koppers Co., Inc., Piston Ring Division in Baltimore, Md.

JOHN M. COLONAS recently became chief engineer with the Commercial Filters Corp. in Boston, Mass.

FRED A. JENNESS has been appointed Detroit automotive sales representative of Clark Equipment Co. He was formerly associated with Timken Roller Bearing Co., Graham-Paige Motors Corp., and Reynolds Metals in engineering, production, and sales capacities.

DONALD A. HUELSKAMP, graduate of Indiana Technical College, is a design engineer with the Hydraulic Press Mfg. Co., in Mount Gilead, Ohio; **THOMAS L. WILE,** graduate of University of Michigan, College of Engineering, is now affiliated with the Linde Air Production Co., Tonawanda, N. Y.; **WAYNE E. SHANNON,** graduate of Cornell University, is development Engineer, American Die & Tool Co., Reading, Pa.; and **WALTER E. YURS,** graduate of Purdue University, is design engineer with North American Aviation, Inc., Los Angeles, Calif.

Members

IRVING L. ROSS is a mechanical engineer with the F. H. McGraw Construction Co. in Hartford, Conn.

T. R. SCHULZ is now connected with Frederick Flader, Inc., North Tonawanda, N. Y., as project engineer.

DENNIS HSUEH-HSI LIU has become assistant manager of the W. Y. Liu Garage & Motor Car Co., in Tsingtao, China. He was previously connected with the National Agricultural Engineering Corp., in Shanghai, China, as an associate engineer.

EDWARD H. BARKER, JR., has been appointed assistant dean of Parks College of Aeronautical Technology of St. Louis University.

DAVID T. MARKS has become design and development engineer for the Cummins Engine Co., in Columbus, Ind. Prior to this position he was connected with the Packard Motor Car Co., Toledo, Ohio.

DAVID BLATTNER is installation engineer for the Caterpillar Tractor Co., in Peoria, Ill.

JOHN C. RASMUSSEN is an engineering trainee at the Goodyear Tire & Rubber Co., in Akron, Ohio.

P. W. LITCHFIELD, has been re-elected chairman of the board and chief executive officer of the Goodyear Tire & Rubber Co. He has been chief executive officer of Goodyear since 1926 and has served continuously as a director of the company since 1906. R. P. DINSMORE was re-elected a vice-president.

WILLIAM C. LITTLE is now an associate with Lang & Lang in Detroit. He was previously Detroit district manager, for 33 years, for the Bearings Co. of America of Lancaster, Pa.

HOWARD B. HASKINS has become executive engineer for the Hudson Motor Car Co. in Detroit.

WILSON G. WALTERS is now a mechanical engineer on product development in the Hawk-Eye Division of Eastman Kodak Co., Rochester, N. Y. Formerly he was a mechanical engineer in charge of engine test, Research Division, Mack Mfg. Corp., Plainfield, N. J.

WILLIAM R. WOOD is now employed with the Chance-Vought Aircraft Co., as a tool designer in Dallas, Tex.

ALBERT D. TRAGER is coach service representative at General Motors Corp., Truck & Coach Division, in Charlotte, N. C.

LOUIS S. KUHN is now sales and service engineer for Automotive Products, Inc., in Portland, Ore.

HENRY C. PARSONS has become senior design engineer at Vickers, Inc., Detroit.

JOHN R. HAHNE, prior to taking his present job as process engineer for Ko-Z-Aire Corp., in Red Oak, Iowa, was time study engineer with the A. O. Smith Corp., Kankakee, Ill.

WILLIAM H. GRAVES (right), formerly executive engineer, has been promoted to a new position of vice-president and director of engineering of Packard Motor Car Co. He is a past vice-president of the SAE. COL. J. G. VINCENT continues as vice-president in charge of engineering and member of the Board of Directors. Vincent is a past-president of the SAE.

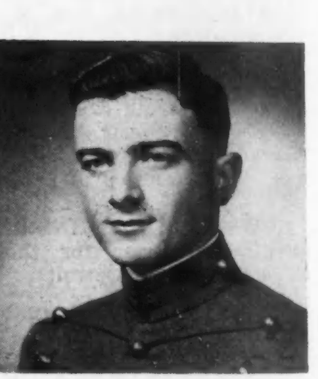
ARTHUR E. SMITH has been named chief engineer of Pratt & Whitney Aircraft Division of United Aircraft Corp. He succeeds to the post left vacant by the recent death of Andrew V. D. Willgoos and is only the second man to occupy the position of chief engineer for Pratt & Whitney Aircraft. Smith has been assistant chief engineer of the Engine Manufacturing Division since 1944.

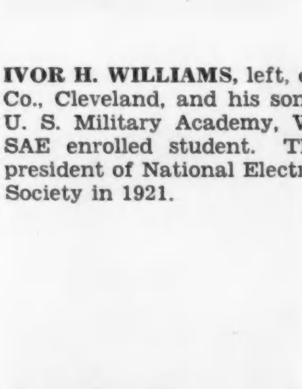
B. R. (BOB) TEREE has been appointed director of engineering, Hydraulic Division of the New York Air Brake Co. He will be located at the operating departments offices of the company in Watertown, N. Y. Teree is an active member of SAE committees and is now chairman of SAE Committee A-6, Aircraft Hydraulic and Pneumatic Equipment. He was previously associated with the Weatherhead Co. in Cleveland and before that with Curtiss-Wright.

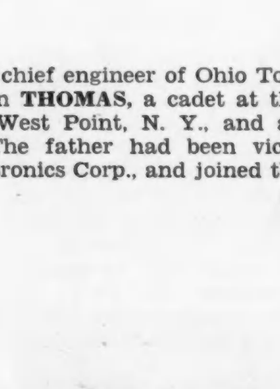
SAE Fathers and Sons . . . . .

EUGENE W. KETTERING, left, chief engineer of Electro-Motive Division, General Motors Corp., LaGrange, Ill., with his father, SAE Past-President **CHARLES F. KETTERING,** a director of and consultant to General Motors Corp. The "Boss" of GM Research for many years served as head of SAE in 1918 and was elected to membership in 1911. His son became a member in 1938.

IVOR H. WILLIAMS, left, chief engineer of Ohio Tool Co., Cleveland, and his son **THOMAS,** a cadet at the U. S. Military Academy, West Point, N. Y., and an SAE enrolled student. The father had been vice-president of National Electronics Corp., and joined the Society in 1921.







EARLE F. TRAISE, left, factory manager, Wilkening Mfg. Co. (Canada) Ltd., and his son **JOHN E.,** an SAE enrolled student who graduates this June, majoring in mechanical engineering, at Lehigh University. During the war the father was factory manager of the company's aircraft piston ring plant in Scranton, Pa., following eight years in the company's sales division in the midwestern states.

CHARLES D. HEALEY, formerly supervisor of motor truck equipment for Air Reduction Sales Co., New York City, has been appointed service manager of the Atlanta, Ga., factory branch of Mack Mfg. Co. He had been a fleet operating engineer with Tide Water Associated Oil Co., New York City, and has been active in Metropolitan Section affairs.

WILLIAM LITTLEWOOD, vice-president in charge of engineering at American Airlines, Inc., has been elected president of the Cornell Society of Engineers. He was graduated in the class of 1920 at Cornell University.

ELEANOR ALLEN, managing editor, SAE Quarterly Transactions, won first prize in the Pre-Salon Group photographic contest of the Rockefeller Center Camera Club (N. Y.) in April. It was Miss Allen's first participation in a photographic contest.

F. E. BREMER is now working in the Aircraft Service Department of Bendix Products Division of Bendix Aviation Corp., in South Bend, Ind.

CHARLES S. FISHER recently purchased the Goodspeed-Detroit Co., distributors of tools and machinery, located at 2832 East Grand Blvd., Detroit. He will continue the business at this location under the same name.

GEORGE E. MALITO has become stress analyst for the Bendix Aviation Corp., South Bend., Ind.

NORMAN F. JONES recently became New England district manager for the Indian Motocycle Co., in Springfield, Mass. Prior to this position he was assistant plant superintendent for G. W. Moore Co., in Waltham, Mass.

HARRY W. LUTZ is now field serviceman for the North American Fibre Products Co., Cleveland.

W. E. JUDD has been appointed general sales manager of the South Wind Heating Equipment Division of Stewart-Warner Corp., Indianapolis, Inc. On May 1 he became South Wind branch manager at Los Angeles, handling sales of home heating equipment and aircraft heating equipment in California.

JOSEPH J. MIKITA has been promoted to director of the Petroleum Laboratory at E. I. DuPont De Nemours & Co., Inc. He joined DuPont in 1946, coming from the Texas Co. where he was associate director of research.

Thompson Products Promotes

WRIGHT

REEVES

BUBB

HILEMAN

Directors of Thompson Products, Inc. have named **J. D. WRIGHT**, vice-president and secretary, to the post of general manager of the company. Four divisional managers were elected to vice-presidencies. They include SAE members: **LEN W. REEVES**, Special Products Division at the Clarkwood Road plant in Cleveland; **HARRY D. BUBB**, Tapco Division at the Euclid plant; and **PAUL D. HILEMAN**, West Coast plant at Los Angeles. President **F. C. CRAWFORD** said that the general manager's post, which he has retained since 1929, was being transferred to Wright as a "further delegation of responsibility in keeping with the continuous growth and expansion of the business." Other changes are: **G. R. MOORE** named manager of the Valve Division; **ANDREW KARABINUS** promoted to assistant manager of the division; **JOHN B. GATES**, of the Detroit sales office, named sales manager for valves; and **JOHN NEWTON** becomes chief engineer. **IRWIN A. BINDER** was named assistant manager of Tapco's Jet Division and **R. E. CUMMINGS** was placed in charge of sales and engineering of jet products. **E. P. RILEY** was promoted to assistant manager of Tapco Parts and Accessories Division. **C. W. OHLY** has been named assistant manager of the company's Cleveland plant and **R. M. WARD** was made sales manager.

GERALD B. HAWKINSON recently became a project engineer at E. I. Du-Pont De Nemours & Co., in Cleveland.

WARNER A. LIPMAN is now working for Cummins Engine Co., in Columbus, Ind.

WALTER N. JAMESON is sales and service manager of Ozalid equipment for the Browning Blue Print & Engineering Co., in Amarillo, Tex. He had been machine shop foreman for Winthrop Motor Co., in Tacoma, Wash.

WILLIAM W. CARY, JR., is now production analyst for the Ford Motor Co. in Dearborn, Mich. He had been liaison engineer at International Harvester Co., Melrose Park, Ill.

J. J. DZIEWONSKI has completed his work in connection with the building and setting up of the Aero Engine Factory for the Turkish Air League in Ankara, Turkey, and is now working in the Production Control Department of the De Havilland Engine Co., Ltd., London, England.

New Vice-Presidents at Redmond Co.

MAURER

BEST

TWEEDY

FOX

Newly appointed vice-presidents at Redmond Co., Inc., Owosso, Mich. include: PAUL H. MAURER, in charge of engineering; PAUL B. BEST, JR., in charge of series motor sales; JAMES W. TWEEDY, in charge of induction motor sales; and WILFRED R. FOX, in charge of marketing research and advertising. Fox, in addition to being vice-president is manager of advertising and marketing research for an affiliated company, Holtzer-Cabot, Inc. of Boston

E. A. ROBERTS, tire engineer in charge of passenger car tire design for the Firestone Tire & Rubber Co., has been assigned to the Detroit area as Firestone resident engineer. Widely known in the Detroit area for his developments in the field of passenger car tire engineering he has presented a number of papers on various phases of this subject before SAE meetings.

EDWARD B. HILL, formerly general sales manager of Gar Wood Industries, Inc., has been appointed vice-president in charge of sales. He is one of the oldest employees of the company, having started with Wood Hydraulic Hoist & Body Co., predecessors to Gar Wood Industries, Inc., as a mechanic helper in 1926.

JOSEPH A. DOYLE has been appointed director of the new Government-Fleet Division of Sun Electric Corp., Chicago, with offices in Chicago. He was previously regional manager of the company in New York. During the war he was overseas as commander of Ordnance Heavy Maintenance battalions in several theaters and retired with the rank of lieutenant colonel in the Ordnance Reserve.

WILLIAM C. WOLD has opened offices at 500 Fifth Avenue, New York City, where he will engage in representation, sales promotion and special projects under the name of William C. Wold Associates. His resignation as New York manager of Consolidated Vultee Aircraft Corp. was announced recently. LaMotte T. Cohu, Convair's president, announced that Wold will provide New York sales representation for Convair on a contract basis.

J. A. PACKARD was recently appointed manager of original equipment sales for the Fuller Mfg. Co.'s Transmission Division in Kalamazoo, Mich. He joined this company as sales engineer in 1942.

Dr. STEPHEN J. ZAND, left, vice-president in charge of engineering, Lord Mfg. Co., Erie, Pa., receiving the citation accompanying the Medal of Merit awarded to him by Secretary of the Air Force W. Stuart Symington March 3 in behalf of President Truman. Dr. Zand, as a colonel in the Air Technical Intelligence, the citation read in part, "contributed immeasurably" to the U. S. Strategic Bomb Survey because of his "initiative, remarkable linguistic abilities, broad knowledge of aeronautics . . . made his services indispensable to our war effort"

OBITUARIES

ALEXANDER STANLEY McARTHUR

Alexander Stanley McArthur, a "founding father" of SAE's Canadian Section, died April 6 as he stepped from a plane in Washington, D. C. He had gone to Washington from Toronto to attend a meeting of the National Association of Motor Bus Operators of which he was a director.

Col. McArthur was a veteran of two world wars, in both of which he served overseas. In World War I, he was gassed while serving with the 35th Battalion, Canadian Engineers and later was code officer with the British Army Detachment in Washington. In World War II, he was promoted to Lieutenant Colonel in charge of No. 2 Canadian Army Ordnance Workshop. Taking this unit overseas, he lost an eye in a jeep accident in England. His three sons also held commissions in the Canadian forces in World War II.

In 1943, he returned to his duties as assistant manager in charge of operating equipment, Toronto Transportation Commission. He had been associated with the city transportation system for 37 years, having been appointed resident engineer with the Toronto Street Railway in 1912.

Col. McArthur was born in Quebec 61 years ago. He was educated in Toronto, graduating from University of Toronto with a Bachelor of Science degree in 1911. He was chairman of the civil engineering advisory committee of the Engineering Alumni Association of the University. He is survived by three sons and by his wife, a native New Zealander whom he met in England while he was convalescing and she was nursing during World War I.

Described by an SAE associate as "a delightful personality whose quiet geniality and constructive collaborative bent were proof against both the slings and arrows of outrageous fortune." Col. McArthur played an important part in SAE from the time he joined in 1926. He was second chairman of the Canadian Section after its founding, was SAE vice-president for Transportation & Maintenance Engineering in 1932, and continued an active member of the Section's governing board to the time of his death.

BION COLE PLACE

Bion Cole Place, who had been associated with the automotive industry since 1913, passed away on March 29. He was 73.

Place was born in Goshen, Ind., and was a widely known inventor. For the past 21 years he was sales engineer for Gagnier Fibre Products.

He was a member of the Ohio bar, and the Detroit Engineering Society.

CALENDAR

Buffalo—May 24

Niagara Falls Power Co. Plant, Buffalo & 16th St., Niagara Falls, N. Y.; tour through the old Edward Dean Adams plant and the main plant of the Niagara Falls Power Co. Dinner 8:00 p.m. at the Red Coach Inn. History of Western New York Industry and the roles played by the plants visited—Fred Koethen, Goetz-Petro-Chemical Co.

Canadian—May 20

Genosha Hotel, Oshawa, Ont.; dinner 7:00 p.m. The New Cadillac Engine—E. N. Cole, chief engineer, Cadillac Motor Co. Golf in afternoon preceding meeting.

Central Illinois—May 16

Jefferson Hotel, Peoria; dinner 6:30 p.m. My Friend the Engine—Stanwood W. Sparrow, vice-president in charge of engineering, Studebaker Corp., South Bend, Ind., and president, SAE.

Chicago—May 10

Knickerbocker Hotel; dinner 6:45 p.m. Meeting 8:00 p.m. Highlights of Friction Material Development—Clyde S. Batchelor, director of laboratories, Raybestos Division, Raybestos-Manhattan, Inc. Social half hour; 6:15 to 6:45 p.m. (Sponsored by Raybestos-Manhattan, Inc., Aluminum Industries, Inc., and Inland Steel Co.)

Cincinnati—May 23

Presidential Meeting. My Friend the Engine—Stanwood W. Sparrow, vice-president in charge of engineering, Studebaker Corp., South Bend, Ind., and president, SAE.

Dayton—May 24

Presidential Meeting. My Friend the Engine—Stanwood W. Sparrow, vice-president in charge of engineering, Studebaker Corp., South Bend, Ind., and President, SAE.

Detroit—May 23

Statler Hotel; dinner (time to be announced). Speaker: John L. Collyer, president, B. F. Goodrich Co. Toastmaster: W. J. Davidson, administrative engineer, GMC.

Indiana—May 25

Indianapolis Motor Speedway; Indiana Annual Race Meeting to be held in the afternoon. Dinner at the Speedway Golf Club. My Friend the Engine—Stanwood W. Sparrow, vice-president in charge of engineering, Studebaker Corp., South Bend, Ind., and president, SAE.

Metropolitan—May 19

Statler Hotel; meeting 7:45 p.m. Problems and New Developments of Hand Brakes—Ralph K. Super, Timken-Detroit Axle Co.

Milwaukee—May 6

Milwaukee Athletic Club; dinner preceding plant trip through Allis-Chalmers Mfg. Co. Atomic Energy—Prof. Daniels.

Northern California—May 16

Engineers' Club, San Francisco, Calif.; dinner 6:30 p.m. Maintenance and Transportation Meeting. Subject and speaker to be announced.

Northwest—May 6

Gowman Hotel, Seattle, Wash.; dinner 7:00 p.m. Cleaning Processes—Valentine Gephart, president, Valentine Co., Motion picture (courtesy Hall Scott Motor Car Co.)

Philadelphia—May 13

Bala Golf Club; dinner 7:00 p.m. Meeting 8:00 p.m. The Early Days of the Automobile—M. J. Duryea.

Pittsburgh—May 24

Edgewood Country Club. May Outing. Golf will start at 1:00 p.m.; Cocktail party and dinner will follow.

Southern California—May 12

Rodger Young Auditorium, 936 W. Washington Blvd., Los Angeles; dinner 6.45 p.m. Developments in Diesel Starting and Fuel Injection Systems—sales engineer, Diesel Division, International Harvester Co.

Southern New England—May 13

Wethersfield Country Club, Hartford, Conn., dinner 7:00 p.m. Annual spring outing and Ladies Night.

Virginia—May 9

Location and time to be announced. Gasoline Economy—W. S. James, vice-president, Fram Corp., East Providence, R. I.

Western Michigan—May 19

Mona Lake Boating Club, Muskegon, Mich.; dinner 6:45 p.m. (Fish Fry). Cylinder Wear—Paul Lane, Muskegon Piston Ring Co.

Williamsport Group—June 6

Antlers Club, Williamsport, Pa., dinner 6:45 p.m. Annual Ladies Night Dinner-Dance.

NATIONAL MEETINGS • 1949

MEETING	DATE	HOTEL
SUMMER	June 5-10	French Lick Springs, French Lick, Ind.
WEST COAST	Aug. 15-17	Multnomah, Portland, Oreg.
TRACTOR	Sept. 13-15	Schroeder, Milwaukee, Wis.
AERONAUTIC and Aircraft Engineering Display	Oct. 5-8	Biltmore, Los Angeles
DIESEL ENGINE	Nov. 1-2	Chase, St. Louis, Mo.
FUELS & LUBRICANTS	Nov. 3-4	Chase, St. Louis, Mo.
ANNUAL MEETING and Engineering Display	Jan. 9-13, 1950	Book-Cadillac, Detroit

Lack of Data on Use Hampers Designers

• Williamsport Group
H. W. Epler, Field Editor

April 4—The imperfection of the designer's knowledge on how equipment will be used **John R. Griffin, Jr.** labeled as the basic problem facing aeronautical engineers. Griffin is manager of engineering, Delaware Overhaul Base, Transcontinental and Western Air, Inc.

In his talk on "Airline Equipment—Its Design and Abuse," Griffin pointed out that a piece of equipment cannot be life-tested in all elements and all variations of its use in the laboratory. The equipment may successfully endure thousands of hours of testing in the laboratory but may break down after 50 hr in an airplane, where unpredictable and unforseeable conditions may affect its operation. In effect, the airplane now becomes the testing laboratory, a rather expensive and hazardous one, particularly if the piece of equipment is vital to the functioning of the airplane, he said.

He cited as practical examples of this problem encountered during the maintenance of TWA's Constellations and C-54s malfunctioning of the cabin air compressor and temperature control and hydraulically operated flight controls, condensation on windows, flame front in engine intake manifold, and skin looseness at the tail section of the fuselage.

This last item is an interesting example, since the skin looseness, evidently caused by tail flutter and stretching of the skin during warmup, did not occur on similar planes used by another airline. It was found that the planes on which this stretching did not occur were warmed up with the flaps down, thereby deflecting the air from the tail section.

From the standpoint of maintenance costs, according to the speaker, the fuel-injection engine with low-tension ignition is better than the carburetor engine at high altitudes.

During the question period, Griffin stated that the engine is the most troublesome part of the plane, but that spark plug replacements have been greatly reduced since conversion to low-tension ignition. Hydraulic lines up to pressures of 3000 psi have given no maintenance trouble. There is no appreciable sheet metal failure and very little fatigue failure in major structure; this is due in great part to preventive maintenance, in which points where failure may occur are carefully inspected periodically.

You'll Be Interested to Know . . . .

ANGLO-AMERICAN COUNCIL ON PRODUCTIVITY meeting in New York March 31 heard J. H. Hunt, GMC, tell the story of "Standardization in the American Automobile Industry"—with special emphasis on the philosophy and accomplishments of SAE standardization work. "One point will stand repetition," he concluded, "standards are of great economic importance to the American automobile industry." The Council was established to further the program of increasing productivity in Great Britain by exchanging views "as to whether there are ways in which United States industry could cooperate in assisting these efforts and to take such steps as are consistent with this program and the similar objectives of the Economic Cooperation Administration." . . . The amount of mechanization behind each American worker (twice that behind each British worker) is recognized by the Council as responsible in large measure for the greater output per man-hour in many American industries. Hunt spoke in response to a request made by the Council to SAE.

Design Objectives Change with Times

• Colorado Group
D. H. Lamb, Field Editor

March 25—"Objectives in truck and bus development are continually being altered as economic conditions change, legislative restrictions are modified, labor makes new demands, prices rise and fall, and competitive pressures vary," stated **Merrill C. Horine**, sales promotion manager of the Mack-International Motor Truck Co.

In expanding this statement, he mentioned that, "the trend of regula-

tion and legislation since just before the war has been toward liberalization of size and weight limitations." Horine also pointed out, in this respect, that throughout the various states a great difference of regulation exists.

By diagram, Horine illustrated the improvements in bogie design through force studies. "Balancing of reactive forces from driving torque and propulsive thrust have resulted in the elimination of 'bogie-hopping'—which also results in better braking and traction."

Horine showed how the development of trucks had to be made with a thought of a proper balance of economy, performance, and power. The balance of these three factors influences greatly the selection of the powerplant. "Diesel," stated Horine, "has only one thing to offer the motor truck—economy of fuel cost. No diesel has been able to offer flexibility up to gasoline standards."

Transmission design, brakes, steering, springing, and load distribution were all discussed. Horine also pointed to the improvements in the comforts afforded the driver. These include such factors as better heating and insulation, greater visibility, lowering of the noise level, more comfortable seating, and convenience of controls and instruments.

High Sulfur Adds to Fuel Problems

• Hawaii Section
Rene Guillou, Field Editor

March 21—Petroleum refiners are being pressed from two sides: the need for fuels of higher antiknock rating and the steady upward trend in percentage of sulfur in crudes from all sources, explained **Charles F. Becker**, automotive laboratory supervisor for Tide Water Associated Oil Co.

Quantity production of high-octane gasoline is going to be difficult to obtain at reasonable cost. Instead, some type of antidetonant injection may be a better solution to the antiknock problem, Becker said, because it would allow both high- and low-compression engines to operate economically on the same basic fuel.

He felt that improved detergency in lubricating oils is the most promising contribution the industry can make to the solution of the sulfur problem.

Also he stressed the importance of continuous, adequate crankcase ventilation. A combination filter and electric blower unit, for installation on a standard breather pipe, was exhibited as a practical means of improving crankcase ventilation at small expense. Ventilation of the crankcase by intake-manifold suction becomes less desirable as the crankcase exhaust becomes more corrosive and gummy; moreover, this method provides least ventilation at heavy loads, when it is most needed, Becker explained.

Engine design for better temperature control he considers of especial importance as sulfur content of the fuel increases, especially avoidance of low cylinder-wall temperatures and provision for control of oil temperature by means of oil-water heat exchangers. Removal of contaminants from the oil by filtering was also indicated as a possibility; filters being designed to remove contaminants with-

Photographed on the occasion of SAE President S. W. Sparrow's presentation of a charter to the University of Illinois Student Branch are: (left to right, front row) T. A. Bissell, manager of SAE Meetings Div., President Sparrow, Student Branch Chairman J. R. Tucker, Faculty Adviser W. L. Hull, SAE Central Illinois Section Chairman R. C. Williams, and (back row) Student Branch officers Sawicki, Pontius, Steier, Giertz, and Corrigan

The Canadian Section Governing Board held a special luncheon in honor of SAE President S. W. Sparrow on the occasion of his visit to speak to the Canadian Section on April 1. At the head table were: (left to right) SAE Councilor Norman H. Daniel, Past SAE Councilor Neil P. Petersen, Past SAE Councilor W. E. McGraw, Sparrow, Canadian Section Chairman Warren B. Hastings, Norman G. Shidle, editor of SAE Journal, Canadian Section Vice-Chairman William W. Taylor, and D. C. Gaskin, Canadian Section Hamilton District vice-chairman

out removing the additives, and being intended not so much to prolong the life of the oil as to maintain it in better condition.

The tropical audience was entertained with movies of field-testing of oils in the snow drifts and subzero weather of a mainland winter and finally by an ear-splitting demonstration of a model jet engine. About the size of a clarinet, this little unit registered a thrust of over 5 lb or better than 10 hp at 750 mph, according to the mathematics of one of the audience.

Air Passengers Will Demand Turbine Power

• Baltimore Section
Herman Hollerith, Jr., Acting Field Editor

March 10—Regardless of relative efficiency of turbine powerplants, the air line passenger of the future will demand turbine type engines because of the smoothness of operation and because they will want to travel in the most up-to-date equipment, **D. Roy Shoults,** vice-president of Glenn L. Martin Co., told members and guests of this Section at a dinner meeting this evening.

In his talk, "Significance of New Turbine Engines to Transport Airplane Design," Shoults compared the relative efficiencies, operating characteristics, suitability for various length of routes, and other operational factors of reciprocating, propjet, and jet prime movers.

He also discussed comparisons of reciprocating, compound, turboprop, and turbojet engines, citing their respective advantages and disadvantages.

Technical Chairman and sponsor of the meeting, Garth Bair, introduced Hollister Moore, SAE headquarters staff, and ranking Glenn L. Martin engineers and executives.

Chairman A. L. Theobold of the Section's reception committee entertained prior to the dinner.

Luxford Discusses Turbosupercharging

• Northern California Section
F. G. Wildhagen, Field Editor

March 28—How the Buchi system of turbosupercharging makes use of some of the energy of exhaust gases and pays off in increased shaft horsepower

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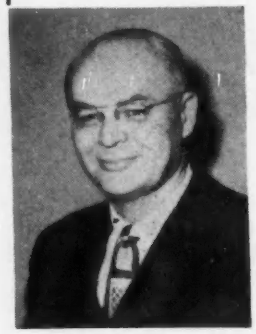
SAE Section



SCHAKEL
. . . . of Indiana

Raymond Anton Schakel's automotive engineering career is the product of neither whim nor accident. Inherent leanings stimulated by environment cast the die in Ray's case.

When Indianapolis was the Motor City, he spent many hours watching Marmon, Stutz, Cole, Premier, American, and Pathfinder road-test cars go by, sometimes getting the thrill of a ride offered by a kind driver. Neighbors Cannon-Ball Baker, 500-mile driver Joe Dawson, and the Speedway furthered this automotive stimulus. Ushering jobs at the 500-mile race kept Ray in close contact with the spectacle for many years.

Appeal of things mechanical and transient headed him for Purdue where, following a brief time in the N.R.O.T.C., he went on to his B.S. degree in Mechanical Engineering, with an automotive major.

From college Schakel went with Nordyke & Marmon Co., spending a year in sales and service school and another year selling Marmons in St. Louis. He then returned to Indianapolis and sold Cadillacs, until going with the Diamond Chain Co. in 1924.

After a year of special chain engineering and promotion, he became Diamond's field engineer out of Pittsburgh, Cleveland, and Chicago for 10 yr. Ray then returned to Indianapolis and for the last 13 yr has been promoting new fields for chain application in general, including automotive timing and transmission drives.

Ray attends the SAE Annual, National Tractor, and Summer Meetings and the ASME Annual Diesel Meeting in the line of duty. Occasionally he takes time off to officiate for the Amateur Athletic Union at swimming meets, a hobby acquired through progeny who have been enthusiastic participants in the sport.

JOHNSON
. . . . of Western Michigan

That geniality and constructive help for which his friends in the foundry field know him, Willis R. Johnson carries over to his Section activities.

Bill was graduated from Muskegon Junior College and then studied at Ferris Institute, where he majored in engineering. In the several jobs he held after graduation Bill gained valuable experience for his first love—metallurgy.

In 1934 he went to Campbell Wyant & Cannon Foundry Co. as melting superintendent. His work in the production end of the foundry has given him the fine background for his present position of sales engineer. In this job Bill has made many friends throughout the industry with his genial personality, his ever-present smile, and his genuine knowledge of foundry and casting problems.

Johnson joined SAE in 1945 and took on the always-important post of program chairman. Because of the fine job he did, Bill was elected secretary of the Western Michigan Section, and the following year, section chairman. He brings a youthful zest to the problems of keeping a young Section on its toes. He has worked hard toward bringing in new members from all corners of the Section by keeping in mind needs of the Section.

In addition to his SAE work, Bill is a member of the American Foundrymens Society, and American Society for Metals, and vice-president of the American Grey Iron Founders' Society.

Not all of Bill's casting problems are encountered in his work. Almost every weekend he can be found casting on the beautiful Pere Marquette River for those elusive rainbow trout. He also has a fine Labrador Retriever that accompanies him in duck hunting on the Grand River, or pheasant hunting at any likable spot in the Section.

—Herman Stapel, Field Editor

Chairmen

These biographies are part of the series on 1948-1949 Section Chairmen

JEUDE

. . . . of St. Louis

A sparkplug in the Busch-Sulzer Brothers Diesel Engine Co. engineering department, Adolph Jeude, began his career with the company in 1915 as a draftsman.

During the early period of his service with Busch-Sulzer, he sandwiched in five years of night school and then took Washington University's extension courses. Still with a yen to study, he took special courses in design and stress analysis with his chief design engineer at the plant.

Jeude spent five years as a designer and two years in Busch-Sulzer's New York office as a sales engineer. Later he returned to the St. Louis plant as a design engineer and was appointed assistant chief engineer in charge of design in 1942. He became assistant chief engineer in 1946.

Except for the two years as New York sales engineer, he has specialized in design developments for stationary, marine, and locomotive diesel engines. During World War II, he did development work on Army and Navy sea-going-tug diesels and cargo-vessel diesels.

After joining SAE in 1945, Jeude served as vice-chairman representing Diesel Engine Activity during his first year in the organization and was the Section vice-chairman for the 1947-1948 season.

Although Adolph pursues his hobby of photography with a seriousness that only an engineer can muster, his ability as a yarn swapper has every listener begging for more. His color movies, which are of professional quality, take up most of his spare time, but he takes a few fishing trips every year—never without his camera.

—A. D. Trager, Field Editor

BROWN

. . . . of Salt Lake

Into his thirty years David Brown has packed lots of automotive experience.

He studied mechanical engineering at the University of Utah, starting in the automotive field to help pay for his education.

He first worked for the Lang Co. as an industrial parts man. Dave then went to the Ford Motor Co., Salt Lake Branch, as a shipping and receiving clerk in the parts department. During the war years he worked for Eimco Corp. as a draftsman and at the Naval Supply Depot, auditing all automotive parts accounts.

In 1942 Dave became associated with Cummins Intermountain Diesel Co. as parts manager. He became a partner in the firm in 1946 and took over management of the sales department.

Color photography takes much of Dave's spare time, although he also likes hiking, golf, and baseball. Dave and Marion Brown have two daughters, eight and five years old.

—Stanley Stevens, Field Editor

was explained by **J. W. Luxford**, field engineer for the Elliott Co.

The Buchi system of turbocharging utilizes the initial high pressure of the gas leaving each cylinder at the beginning of the exhaust stroke by the proper proportioning of exhaust manifolds and the turbine nozzle area, Luxford explained. This energy is used to drive the turbine. After the initial surge of the exhaust gas from the cylinder, the pressure in the exhaust manifold drops nearly to atmospheric, while the air pressure from the blower remains considerably higher (3½ to 4½ psi gage).

The inlet valve is then opened, before the exhaust valve closes, mixing cool air with the hot exhaust gases in the manifold. After the exhaust valve closes, the inlet valve remains open during the beginnning of the compression stroke, and the cylinders are filled with relatively cool, fresh air under pressure. In order to prevent interferences between the gas impulses, and to prevent exchanging exhaust gas with other cylinders during the scavenging period of a given cylinder, separate manifolds from one, two, or three cylinders are conducted to separate banks of the nozzle in the turbine.

Generally, a straight reaction turbine stage is most efficient for this type of service. The usual mechanical problems of close clearances and axial rotor thrust are minimized in these turbines due to the low pressures encountered.

New Alloy Used

To produce a successful turbocharger, it was necessary to develop metals that would stand extremely high temperatures at relatively high rotative speeds, such as a speed of 21,000 rpm for an 11-in. diameter wheel at a temperature in excess of 1000 F. Luxford reported that a stabilized austinetic stainless steel, 19-9 WMO, has been successful. This metal contains chromium, nickel, molybdenum, tungsten, titanium, manganese, carbon, silicon, and columbium. The metal is very ductile and rather difficult to machine; it requires very sharp tools. The creep rate is appreciable for the first few hundred hours of operation, but after that, it stabilizes and reduces almost to zero.

Turbocharged engines show a very flat fuel rate curve over a wide load range, Luxford said. Economies of 25% have been shown at one-fourth-load operating conditions. Full-load fuel rates are comparable to nonturbocharged engines. Naturally, exhaust-gas turbocharged engines show a very definite economy in fuel rate over those supercharged by other means, due to the elimination of the parasitic power requirement of the blower. While it is true that the turbine exerts a slight back pressure on the engine (about 2¼ psi), this slight reduction in engine

output is more than offset by the load-carrying ability of the engine, due to the cool dense charge of air in the engine cylinder provided by the turbocharger, according to the speaker.

A well designed engine can be turbocharged to approximately 50% increase in output, he said. There have been a number of instances where the turbocharged output of the engine has been increased as much as 60% or 70%. Naturally, there have been changes in the engine itself to accomplish these high outputs. Application of the turbocharger necessitates a thorough study of the engine, and, although in many instances turbocharging itself did not provide the additional power output, it was in a measure responsible for more thorough and complete engineering and modernization of the engine to be turbocharged.

The initial application of turbochargers to 4-cycle engines resulted in bmep's of approximately 115 to 120 psi. While this has been a conservative figure, it is not uncommon to see test floor results in excess of 160 and 170 psi bmep. In all probability, the 4-cycle engine of the future will show considerably higher bmep's than those considered reasonable today, Luxford predicted.

Texas Welcomes Chance Vought

• Texas Section
Floyd Patras, Field Editor

April 8—Texas Section welcomed the engineering and production people of the Chance Vought Division of United Aircraft Corp., who are in the process of moving plant and personnel from Stratford, Conn., to Dallas.

In return, **John J. Hospers**, sales manager of Chance Vought and vice-president of United Aircraft Service Corp., reviewed the history of the division and its designs.

On display were models of various Chance Vought aircraft.

Zero Reader Aids Instrument Landings

• Dayton Section
J. E. P. Sullivan, Field Editor

March 22—With the new zero reader, an airplane pilot can set his compass heading and altitude, then maintain them by keeping vertical and horizontal cross pointers centered at zero on a small dial, explained **F. Glenn Nesbitt**, liaison engineer with the Sperry Gyroscope Co.

Greatest advantage of the zero reader lies in its use in making instrument landings in inclement weather, according to Nesbitt. He added that by means of instrument landing systems, 56% of our commercial airlines can operate when landing ceilings are between 200 and 400 ft.

The zero reader does not replace the pilot, but it relieves him of constantly checking the myriad instruments while in flight, Nesbitt said. Zero readers will be available commercially early in 1950, he reported.

Sparrow Visits Western Michigan

• Western Michigan Section
H. F. Stapel, Field Editor

March 17—SAE President **Stanwood Sparrow** was honored by 150 members and friends gathered at the annual dinner meeting.

Sparrow remarked that each manufacturer would like to build the most powerful car on the road, but its cost would prohibit prospective buyers from owning it. Therefore, engines are built to sell in competition.

Master of ceremonies for the meeting was Paul Lane, who was introduced by Section Chairman W. R. Johnson. R. C. Sackett, SAE staff member, presented certificates to former chairmen Paul W. Fuller and G. Waine Thomas. Thomas having fulfilled Fuller's unexpired term when the latter moved to Chicago.

Fractures Give Clues To Causes of Failure

• Central Illinois Section
R. V. Larson, Field Editor

March 28—Appearance of the fracture of a part that has failed in actual service can provide clues to the causes of failure, **R. E. Peterson** proved.

Peterson is head of the mechanics division of the Westinghouse research laboratory.

Location and appearance of the small portion of the fracture which breaks last is the primary clue, he explained. Large stress concentrations tend to restrict the fatigue fracture to the areas in which the stress concentrations occur.

Failures along long erratic paths indicate that failure is due to something other than high stress concentration.

Direction and rate of propagation of the failure depend on the stress distribution across the part. The fact that a crack has appeared does not necessarily mean that the part has

failed. Peterson warned. The rate of propagation may be such that many hours of serviceable life remain. Periodic inspections will disclose data on rates of propagation.

Fracture appearance often indicates the type of load causing failure. With practice, fractures produced by bending stresses can easily be distinguished from torsional-load failures.

Stress rupture is characterized by separation of the piece along grain boundaries, rather than across grain boundaries as in true fatigue failures. Stress rupture, like fatigue failure, is a function of time and produces failure at loads much lower than the ultimate strength of the material. It occurs in parts subjected to a steady unidirectional load. The break commonly occurs at elevated temperatures and without much distortion of the cross-section. In certain materials, stress rupture occurs also at room temperatures.

Variations between service and laboratory failures Peterson ascribed to two differences: Laboratory tests are almost universally run at one steady stress level, with a consequent lack of definite beach marks on the fracture. Specimens are shaped so that stress concentrations are lacking.

Reese Reviews Brake Problems

• British Columbia Group
J. B. Tompkins, Field Editor

March 16—Brakes used on commercial vehicles must have adequate capacity for the weight and speed of the vehicle, cautioned **Paul J. Reese,** automotive application engineer of the Wagner Electric Corp. of St. Louis.

Chairmanned by H. L. Hinchcliffe, operations manager of Shell Oil Co. of B. C. and an SAE member since 1933, the second meeting during March of this group heard Reese cover the complete brake field in a paper entitled "Automotive Brake Problems."

"Skidding of wheels is the factor which limits the maximum torque that the brakes can deliver for the maximum anticipated weight on the wheels and thus limits the stopping ability of the vehicle," said 20-year veteran of Wagner Electric service.

"During deceleration there is a transfer of weight from the rear to the front wheels, due to the force of inertia that is equal to the product of the braking force at the ground level and the ratio of the height of the center of gravity to the wheelbase."

At high rates of deceleration this weight transfer is large enough to reduce materially the braking ability

without sliding rear wheels, Reese declared. A perfect braking system should vary the brake effort distribution with the dynamic weight distribution changes. Since this is not feasible and as all wheels can develop their maximum friction only at one rate of deceleration, the solution to the problem of distributing brake effort to suit various body types and possible load distribution must be a compromise.

With present design trends toward increased weight on front wheels, the 60% brake distribution on these wheels must remain to maintain present performance. Smaller wheels, wider rims, and larger section tires increase cooling problems and limit brake drum diameters, Reese reminded his audience. Some tests indicate a lower skidding coefficient of friction for synthetic than natural rubber tires.

"During braking action, kinetic energy of the moving vehicle is converted, through friction between the lining and the drum, into heat energy. Drums have been known to develop temperatures as high as 1400 to 1500 F momentarily in severe service," claimed Reese. "But it is only the metal on the inner surface that is raised to this very high temperature. Internal strains produced by this unequal expansion tend to cause the surface metal to check, with the amount of checking dependent on the physical properties of the drum metal."

Brittle metals, the brake authority said, will check very readily. Each repetition of the same condition will deepen the check until it eventually becomes a crack through the drum section. Brake drum materials must withstand brake shoe pressures great enough to flex the drum several thousands of an inch out of round at ordinary temperatures. But the flexing or distortion of the drum, within certain limits, may increase the torque output of a brake over that obtained from more rigid drums.

"Brake drum heat dissipation abilities can be increased by adding cooling ribs which permit more drum area to effectively contact the surrounding air or by causing more air to circulate over the drum surface by coring ventilating holes in the brake drum dish," said Reese.

"New materials may be found for brake drums that will extend their life and usefulness. Experiments are being conducted with hardened alloy iron drums, aluminum drums with cast-iron liners or with sprayed steel liners, cast-iron drums with copper cooling fins or aluminum cooling fins."

Since brake output depends entirely on energy produced from the friction of the brake lining and the drum, Reese stressed the importance of keeping grease, water, dust, and other foreign materials away from the lining. Of the heat produced, 75% is developed during the first half of any deceleration. This, coupled with the brakes' job of overcoming the force of gravity, is the reason why extremely high temperatures occur on winding, mountain grades, Reese said.

Heat Cuts Brake Torque

"Drums expand and surface temperatures reach very high values during brake application," the speaker claimed. "The first result is that the coefficient of friction of most linings decreases to a lower value at elevated temperatures which reduces the torque exponentially. A further result not generally appreciated is that the shoes heat more slowly than the drum. The radius of the shoe is less than the radius of the drum. This causes an additional reduction of torque because of reduced shoe contact."

When severe braking is continued any considerable length of time with the brakes hot, the surface of the lining will be burned and will wear at a very rapid rate to the radius of the enlarged drum. When the brakes have cooled, said Reese, the radius of the shoes will be larger than normal. So will the pressure at the toe and heel. Self actuation will be abnormally high until the linings have worn to the radius of the cool drum.

Brake lining used should be adapted to specific jobs, he reported. Linings differ widely. Some provide greater friction than others with the same pressure against the drum. Some "fade" more and lose friction during long, high speed applications. Some have a greater tendency to glaze and thus lose frictional values.

"Solving the problem of selection," he advised "the most suitable brake lining . . . requires dynamometer and road tests conducted according to a definite procedure which permits a direct comparison of the following:

1. Friction—throughout the life of the lining, hot and cold.

2. Fade—a general description of the reduction of brake effectiveness caused usually by the development of critical temperatures in the brakes.

3. Build Up—a general description of a temporary increase in brake effectiveness caused by an increase of temperature in the brake lining or resulting from a reduction in rubbing speed.

4. Glaxing tendency.

5. Drum wear.

6. Life."

West Congratulated on Inventive Courage

• Southern California Section
R. V. Lindberg, Field Editor

March 17—Speaking as an easterner, **Merrill C. Horine** congratulated the engineers of the west on their courage and unwillingness to shy away from a good idea because it is unconventional or because someone tried it once and did not succeed.

He acknowledged the debt which the truck and bus industry owes to the Pacific Coast for pioneering efforts along such lines as multiwheeled vehicles, multispeed transmission, diesels in trucks, truly big gasoline engines, and the light-weight spoked wheel.

He added that sometimes manufacturers are baffled at the rapidity with which western engineers change their minds, reminding them that no sooner had they persuaded manufacturers that the open-spider spoked wheel was preferable to the sheet-steel disc type than they reversed themselves and demanded disc wheels. He asked for more objectivity in specifications.

Horine, whose paper was titled "Objectives in Truck and Bus Development," is sales promotion manager of the Mack Mfg. Corp.

April 7—**Thomas S. McCrae**, assistant director of engineering, Allison Division, GMC, delivered his paper "Production Problems of Turbojet Engines," which he presented at the SAE Annual Meeting in Detroit last January. T. N. Baker, district representative of Allison Division was technical chairman.

STUDENT NEWS

Fenn College

Heralded as the greatest automotive development since tetraethyl lead, the Vitameter, a device for adding octane numbers to gasoline, was the main topic of discussion by Arthur C. Saxton on March 25. Saxton is sales manager of the Vitane Division and assistant to the vice-president of the Thompson Vita-Meter Corp.

The Vitameter first came into prominence during World War II when it was employed to boost octane numbers for gasoline used in fighter planes. Since then it has been adapted to usage in automobiles, and its 85% methyl alcohol-15% water solution has greatly increased peak power values in high compression.

Basically an auxiliary carburetor, the Vitameter is most often mounted between the carburetor and the manifold, or in front of the radiator. Since it raises the octane number by 12 or 13, the Vitameter is particularly advantageous in fleet operations, where low-octane fuels usually prevail.

Saxton emphasized the fact that the Vitameter will be a vital necessity for the automobiles of tomorrow, inasmuch as the trend is toward high-compression engines, which require gasoline with a greater octane number.

—Robert L. Pappas, Field Editor

California State Polytechnic College

Transportation costs can be considerably reduced when preventive maintenance is properly applied, James A. Hodges, zone fleet service manager, Chevrolet Motor Division of General Motors Corp., said at the **Feb. 23** meeting.

There is no one program for carrying out preventive maintenance. Value and costs must be closely estimated in order to prevent a tendency toward installing a system which would be too expensive for particular operating conditions. Important factors to be considered in any program are proper methods of lubrication and types of gasoline.

—R. B. Canning, Field Editor

Oklahoma A. & M. College

Duties of a sales engineer in the oil industry—that of a consultant engineer serving in marketing areas with salesmen of that district—were explained by Enos W. Cave of Continental Oil Co., in his talk "Sales Engineering and Its Application to the Oil Industry."

A good sales engineer in the oil industry, according to Cave, must have an adequate engineering education, some experience with fuels and lubricants work, an ability to meet people, sound knowledge of basic principles, an ability to "trouble shoot", a way of convincing customers that he knows what he is talking about, and the ability to handle and learn from complaints.

—M. L. Ball, Field Editor

University of Illinois

This SAE Student Branch received its charter from SAE President Stanwood Sparrow at a meeting on March 23.

Sparrow presented his talk "My Friend, the Engine." Chairman James R. Tucker introduced Prof. W. H. Hull, who introduced Sparrow; Dean Stanley H. Pierce, who expressed his satisfaction in the growth of student branches of national professional societies, and Thomas A. Bissell, manager of the SAE Meetings Divisions.

—G. W. Sawicki, Secretary

Special Instruments Aid Engine Refinement

• Southern New England Section
R. E. Johansson, Ass't. Field Editor

March 10—Involved in the refinement of the R4360 Wasp Major engine are many special-purpose instruments, which **Robert E. Gorton**, project engineer in charge of instrumentation at Pratt & Whitney Aircraft detailed.

Pickups most commonly used for vibration are electromagnetic generators. Gorton explained that a permanent magnet is suspended on very soft springs to remain stationary in space as a seismic mass. A coil attached to the vibrating member is forced to oscillate through the field of the stationary magnet, generating an alternating voltage. Vibration is measured from variation in voltage.

Strain gages are also used for vibration studies, he added.

For measuring cylinder pressure and fuel, oil, and air pressures, various pressure pickups are used. The simplest is the balanced-pressure pickup, which is used as a standard for calibration of the others.

It is used for making plots of pressure versus crankshaft angle. Where plots of pressure versus time are required, the capacitance pickup developed by General Motors is applied, Gorton said.

High-speed movie cameras and stroboscopes are used to observe phenomena which cannot be observed at their actual rate of occurrence, he reported.

Bolles Discusses Electric Equipment

• Pittsburgh Section
Murray Fahnestock, Field Editor

March 22—Speaking on automotive electric equipment, **J. H. Bolles**, chief engineer of Delco-Remy, said that starting motors vary from small 1-hp units used on passenger cars to 14½-hp units used on large gasoline and diesel engines.

Recent improvements in starting motors include welded commutators, better impregnation of windings, and alloy switch contacts to prevent burning, Bolles reported.

A few years ago, 10-amp generators were adequate, but now 30-40-amp generators are standard, he said.

Although electronic ignition systems have been in experimental operation, he does not expect that they will be in production for some time because engines will not require them. He believes that in d-c machines commuta-

tion and brush life can be greatly improved by incorporation of interpoles.

ILS Useful Down Only to 200-300 ft

• Wichita Section
R. M. Harmon, Field Editor

March 17—There is no satisfactory blind landing system available today which the airlines can use without risk for making actual runway contacts under zero-zero conditions, according to **R. C. Ayres**, manager of radar, radio, and electrical departments of Trans World Airline.

The present ILS devices in use are usable down to 200 or 300 ft ceilings. However, the airlines are not yet willing to rely on ILS at these ceilings at some airports, Ayres said.

He pointed out also that neither GCA nor ILS systems are considered sufficiently reliable to be depended on alone, but either one used as a back-up system or a combination of both should lower the usual ceiling limits of 200-300 ft now in use.

To utilize this improvement in approach, it will be necessary to have better approach and runway lights to assist the pilot in the final phases of landing, he said.

A movie on ditching procedures for overseas aircraft was shown, and **Ray Bonous** of the Wichita Chamber of Commerce talked on the growth of Kansas.

Airline Airplanes Improve with Age

• Mid-Continent Section
W. F. Ford, Field Editor

March 18—Airplanes kept up to date by modification and overhaul in accordance with CAA, manufacturers', and airlines' standards never wear out.

In fact, said **Otto Kirchner**, director of engineering at the Tulsa maintenance depot of American Airlines, all airplanes operated and maintained by scheduled airline operators improve with age.

Kirchner explained that over 300 improvements are scheduled for the DC-6. They will be made at the Tulsa depot.

An airlines airplane flown 8 hr per day must support over 100 families, pay for its own operation, contribute to the maintenance of all ground equipment and installation, pay for itself, and contribute to the down payment of a replacement airplane, as well as make some profit, Kirchner revealed.

In describing airlines safety provisions, he disclosed that each airplane carries over 2800 lb of equipment added for safety, which costs over $80,000 in potential payload.

Perriguey Answers "Why Drain Oil?"

• Buffalo Section
Benjamin Fuente, Field Editor

March 29—**W. G. Perriguey**, Esso Standard Oil Co. answered the question he asked in the title of his paper "Why Drain Oil?" with the statement that when oil condition deteriorates, engine condition deteriorates.

Going further, he said that the time

to change oil is when the oil condition becomes poor. He recommended to commercial operators that they use used-oil analyses to determine when oil condition requires oil change.

Operators of single passenger cars, who could not stand cost of used-oil analyses, he urged to follow conservative oil drain practices, pointing out that the cost of extra oil is small in comparison with the cost in reduced engine life or performance that may result from over-extended drain intervals.

Geschelin Analyzes 1949 Passenger Cars

• Syracuse Section
W. F. Burrows, Field Editor

March 18—At a joint meeting of the SAE and Steuben Chapter, New York Society of Professional Engineers, **Joseph Geschelin** analyzed 1949 passenger cars, emphasizing new features such as styling, automatic transmissions and their future exploitation, and new engines such as the Cadillac and Oldsmobile high-performance V-8's.

Also discussed were probable developments in engines and automatic transmissions.

Baffles Stabilize Gas Flow in Burners

• Southern New England Section
A. M. Watson, Field Editor

Feb. 10—One of the means **Prof. Glenn C. Williams** described for sustaining combustion in gas turbine powerplant combustion chambers is introduction of baffles to stabilize flow. Williams is associate director of the fuels research laboratory at M.I.T.

Baffles at the forward end operate on the so-called "cold-density" gas masses. Other baffles downstream of the ignition point operate as mixing devices on hot-density gases.

Williams explained that the undesirable pressure loss these baffles incur, due to their stabilizing effect, the momentum change, and the mixing effect, is required for acceptable thermal efficiency.

Present at the meeting were SAE President Stanwood Sparrow and John C. Hollis of the SAE staff. Prof. L. C. Lichty of Yale was technical chairman.

Sheppard Diesel

Cont. from p. 70

out of fuel it merely shuts down the engine, avoiding engine runaway.

The lower two-thirds of the spherical combustion chamber is inserted into the bottom surface of the head and locked with a latch pin and screw. The chamber is connected to the cylinder by three passages.

This combustion chamber and the fuel injection system will burn any fuel from kerosene to light crude with clean exhaust. Although not recommended, it will also handle cottonseed, linseed, soy bean and peanut oils.

The Sheppard aircooled diesel is, we believe, the first commercially successful aircooled diesel powerplant.

One of these has been running 11,500 hr rated 5.4 hp at 2000 rpm without attention to valves, rings, bearings, or fuel pump. (Paper "Features of the Sheppard Diesel" was presented at SAE Annual Meeting, Detroit, Jan. 14, 1949. This paper is available from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Passenger Comfort Urged In Transport Plane Design

Based on papers by

JOHN J. HARRINGTON
Northwest Airlines, Inc.

and

WALTER DORWIN TEAGUE
Designer

MODERN airliner design must satisfy both psychological and physiological needs of passengers to exploit fully advantages of air travel and make it fully competitive with other forms of transportation.

For bodily comfort, the passenger wants:

1. A good seat,
2. To get up and stretch his legs,
3. Freedom from noise, and
4. Proper heating and ventilation.

He will be at ease mentally if:

1. The color scheme is right,
2. Surroundings are conducive to a sense of security.

On seating comfort, studies indicate that certain conventional seat construction ideas carried over from other fields should be discarded; for example, notion that a chair should consist of separate seat and back cushions supported by a frame hampers invention. The designer should approach the problem from a fresh, unbiased viewpoint.

A new type of seat construction and cushioning, extremely light in weight and with great strength, is fully adjustable from the upright to fully reclining positions. And the leg room at 36 in. equals that provided by current seating at 40-in. spacing.

Through a fresh approach such as this it may be possible to eliminate need for space-consuming berths on even transoceanic flights.

In addition to maximum seating comfort, on long flights the passenger should be able to leave his seat, stretch his legs, and change his environment. The small rear lounge of the DC-6 and the lower-deck buffet or cocktail lounge of the Stratocruiser make this possible. There is no more effective way of relieving monotony on a 10 or 12-hr flight than to go below deck in a different setting, seeing different faces, occupying different seats, and being able to order a drink.

Noise also affects physical comfort. Frequency as well as intensity or loudness must be considered if sound sensations are to be minimized.

For successful sound insulation, all cracks and openings in the insulating structure should be avoided. The smallest cracks nullify the best soundproofing treatment. Sound entering a narrow aperture can produce almost the same sensation of loudness as an open window.

Sealed openings around hatches, doors, windows and ventilators must be maintained in good condition. As much as possible of the airplane structure's area should be acoustically treated and untreated metal surfaces kept to a minimum.

While the body can adjust itself to noise or silence and brilliant light or darkness (although its efficiency may be impaired), the body has no adequate defense against extremes of heat and cold.

Airplane designers can provide an environment in which every passenger —not just the ones nearest the heating or ventilating source—can be comfortable. We know people's thermal needs and have the technology for serving them.

Because of passenger inactivity (the body eliminates a minimum of Btu when at rest), constant temperatures —not less than 70 F—should be maintained. Recently developed combustion-type heaters make it possible to distribute great quantities of heat uniformly throughout the passenger cabin.

This is done partially with side-wall heating.

Air in an aircraft must be changed often because of cigarette smoke and food odors in the atmosphere. This assists rather than hinders the heating problem because engineers use ram air directed through the combustion-type heaters to provide thermal requirements.

The mind as well as the body must be at rest if the passenger is to derive complete satisfaction from air travel. Certain factors in airplane design and flying tend to create apprehension in the minds of passengers at take-off. But shortly after take-off, engine noise decreases and the sensation of speed disappears, and the passenger loses his fear of height. All design details which can contribute to the reduction of stimuli causing these sensations within the aircraft must be incorporated.

For example, fine workmanship in equipment not only is attractive, but gives passengers confidence in the airplane itself. It is psychologically reassuring to the passengers. For the same reason most meticulous design study should be given to all minor appointments of lavatories, galleys, and seat conveniences.

Color also must be used as a calming influence for psychological reactions in passengers. We have learned that lighter shades of gray tend to calm fears of height and also curb tendency toward claustrophobia. There are other theories on color schemes and fabric textures aimed at pleasing, reassuring, and satisfying the passenger, many of which are a step in the right direction.

For example, one school of thought has it that all major tones should be kept on one side of the spectrum—in either the cool or the warm range. Most prefer cool colors in airplanes. Light, cool tones contrast in a less unfriendly way with what one sees through the window of an airplane. They are more restful because they require less eye accommodation. Like Nature, the designer should use cool colors for large areas and hot colors for smaller accents.

As airplane size and flight duration increase, efforts aimed at passenger comfort must be accentuated. Achieving this objective will no longer yield unfavorable comparisons between air travel and slower, but at present more luxurious forms of transport. (Papers "Interior Treatment of Aircraft" by Harrington and "The Design of Airplane Interiors" by Teague were presented at SAE Annual Meeting, Detroit, Jan. 14, 1949. Each of these papers is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ each to members, 50¢ each to nonmembers.)

Faulty Refacing Leads to Valve Ruin

Based on paper by

J. E. LACKNER

The Texas Co.

SING a dull or loaded grinding wheel to reface valves can produce grinding checks on the valve face, which will lead eventually to valve burning and failure.

Microscopic examination reveals that these grinding checks are actually hairline cracks. They result apparently from intense local heating and subsequent cooling at the region of contact between the wheel and the valve face. Refacing operations not sufficiently

severe to blue the valve face can be severe enough to initiate cracks.

A dull or loaded wheel accentuates the temperature fluctuation.

The tiny cracks become foci for stress concentrations. High-temperature operation and temperature cycling enlarge the cracks, and deposits wedge into them. At high temperatures, the deposits burn. Eventually, burning washes out gutters across the valve face.

The cracks continue to grow until finally sections of the valve burn and break away.

Good refacing practice demands good equipment, skillfully operated. Concentricity of valve face with stem should be held within 0.003 in. total runout. Valve face angle should match seat angle exactly, unless a ¼ to 1 deg angle is being maintained between face and seat as prevention against guttering. (Paper "Factors Influencing Life and Performance of Internal Combustion Engine Valves" was presented at SAE Oregon Section Feb. 18, 1949.)

New Members Qualified

These applicants qualified for admission to the Society between March 10, 1949 and April 10, 1949. Grades of membership are: (M) Member; (A) Associate; (J) Junior; (Aff.) Affiliate; (SM) Service Member; (FM) Foreign Member.

British Columbia Group

Fred Marrington (A).

Buffalo Section

Peter C. Bila (J), James S. Thompson, Jr. (A).

Canadian Section

Stanley Charles Malcom Ambler (M), Louis R. Clinton (A), John K. E. Cox (A), Everett Richard Smith McLaughlin (J), Lt. Col. Herman G. Morrow (A), Roy H. Sjoberg (M), D'Arcy John Sweeney (J), James Joseph Wall (A).

Central Illinois Section

John C. Eirk (A).

Chicago Section

Richard D. Abelson (J), Johnson S. Davis (A), Leonard R. Fergin (J), Brian M. Gallagher (J), Emil H. Hahn (J), Elmer F. Heimbuch (M), Charles H. McDonnell (J), Jon Renna Morlen (J), Peter A. Mueller (A), Chester E. Palmer (M), Clarence Hugo Patrie (M), C. Robert Powers (A), John A. Toth (M).

Cincinnati Section

Robert H. Jahnke (A), Perry E. Schmidt (A).

Cleveland Section

J. G. G. Frost (M), Floyd Harold Lawson, Jr. (J), Ralph William McCarty (J), Robert Allen Pejeau (J), Carl G. Russell (M), Julius J. Sabo (J), William F. Stoermer (J), Raymond F. Upp (J), Frank H. Webber (A), Bernard A. Wieczorek (J).

Colorado Group

Frederick D. Wilson (M).

Dayton Section

Rodger E. Geller (M), William Bernard Smith (J), William L. Tenney (M), Marvin L. Yeager (J).

Detroit Section

Rufus P. Austin (A), Wayne A. Brooks, Jr. (J), George Arthur Brown (A), William H. Coatney, Jr. (J), Charles R. Cole (SM), George W. Conover, Jr. (M), William G. DeKam (M), Lawrence J. Easterday, Jr. (J), Harry E. Erwin (J), Edward V. Francoise (A), Mark J. Garlick, Jr. (M), Thomas Maxwell Greene (M), James Joseph Griffin (A), James Jack (M), Edward T. Kantarian (A), Wayne C. Keith (J), Howard C. Kellogg (M), Uolevi L. Lahti (A), Gordon I. Lyman (A), Alex C. Mair (J), Donald H. McPherson (J), W. Penn Norris (J), John B. Ready (M), Robert Warren Schleicher (J), A. J. Steger (A), William Kerby Steinhagen (J), Robert Ullman (M), Lloyd R. Vivian, Jr. (J), Edward H. Wallace (M), Earl R. Wilson, Jr. (A), George Wilson (M), Rex Wiltse (A), Stephen Woods (J).

Hawaii Section

Drury Adams (A), Robert M. Bailey (A), L. Earl Derr (A), Everette Wayne Evans (A), Sueo Hayashida (A), Raymond A. Macaulay, Jr. (A), Harry H. Smyth (A), Arthur N. Wayne (A).

Indiana Section

Charles Marion Keepers (M), Fred J. Schmidt (M), Phineas R. Youngs, III (A).

Kansas City Section

Glover Lee Williams (A).

Metropolitan Section

Clarence Milton Anderson (SM), P. A. Chieri (M), Charles Arnold Kalman (A), J. Lawrence Kess (A), David J. Long (A), Adolph Losick (J), Theodore A. Nerlinger (A), Roy C. Norton, Jr. (M), Erwin Rausch (J), Everett H. Schroeder (M), Wendell E. Simpson (A), Thomas George Zsembik (J).

Mid-Continent Section

Howard H. Belew (A), L. B. Goodson (M).

Milwaukee Section

Ivan P. Baxter (J), F. Burrows Esty (M), Charles L. Spraker (J), George Q. Wallace (M).

Mohawk-Hudson Group

Donald J. DeMond (A), Wilfred E. Kenny (A), Clair L. Pepperd (A), Harry L. Ross (A), Charles William Wyld (A).

New England Section

Benjamin R. Chase (A), Richard T. Cole (J), Ralph G. Fritch (J), Albert G. Gunther (A), Sherman L. Whipple, Jr. (A).

Northern California Section

Herman V. Boley (M), Carl E. Bremer (A), Frank S. Bryan (A), R. H. L. de S. Illesinghe (M), Harrison W. Sigworth (J), Clair N. Wikander (M).

Northwest Section

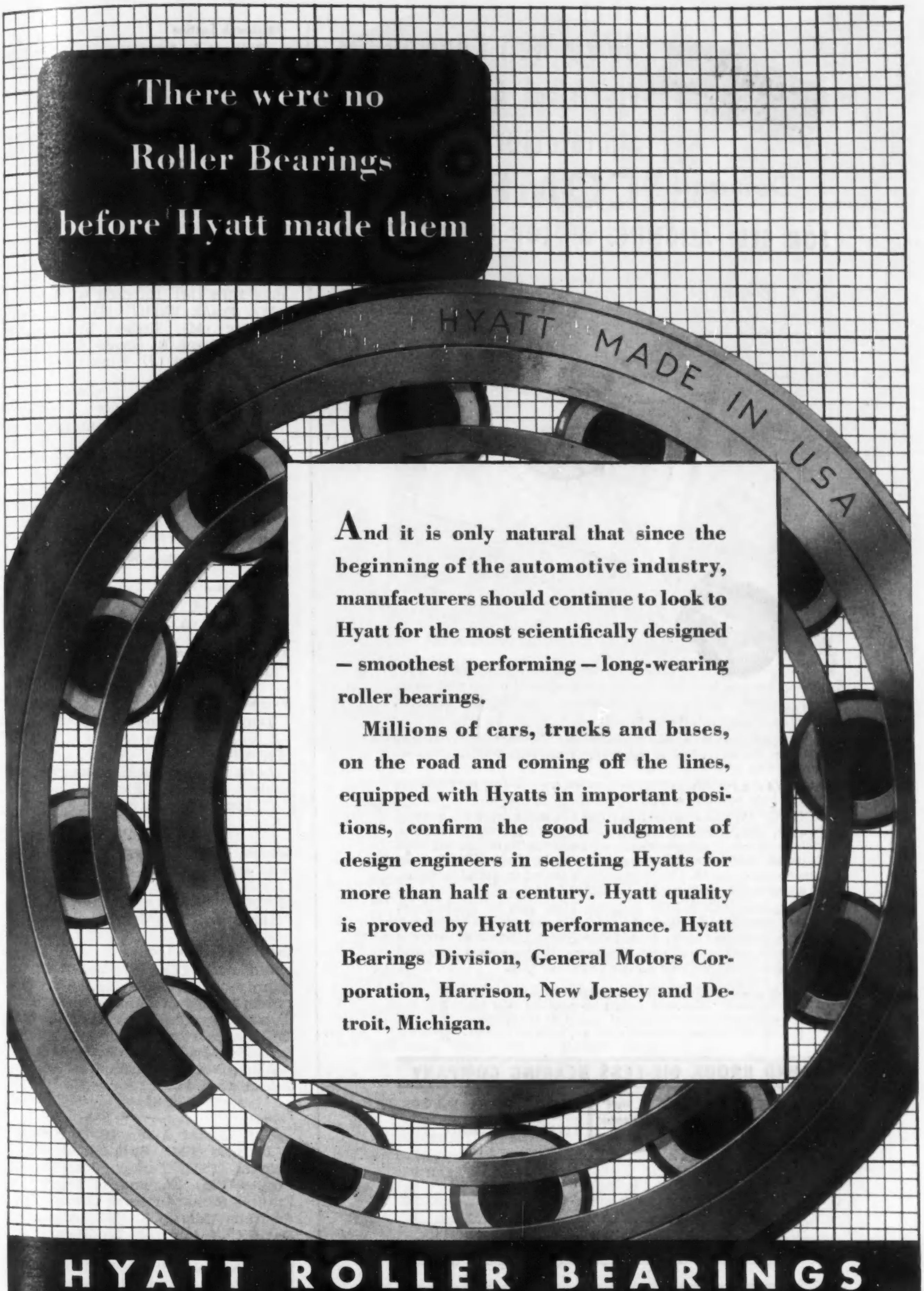
David E. Basor (A), J. E. Cannon (A).

Oregon Section

Erwin W. Eatock (A).

Philadelphia Section

Wilfred H. Bernard (M), John P. Casserly (A), Vincent S. Glowacki (J), George B. Hill (M), Paul E. LaFrance (M), Leon Prager (A), Harold Fielding Watson (M).


There were no
Roller Bearings
before Hyatt made them
HYATT MADE IN USA
And it is only natural that since the beginning of the automotive industry, manufacturers should continue to look to Hyatt for the most scientifically designed — smoothest performing — long-wearing roller bearings.
Millions of cars, trucks and buses, on the road and coming off the lines, equipped with Hyatts in important positions, confirm the good judgment of design engineers in selecting Hyatts for more than half a century. Hyatt quality is proved by Hyatt performance. Hyatt Bearings Division, General Motors Corporation, Harrison, New Jersey and Detroit, Michigan.
HYATT ROLLER BEARINGS

Pittsburgh Section

William L. Armstrong (A), Per Eric Eckberg (J), Hugh J. McGinn, Jr. (A), Edward G. Mosher (A).

St. Louis Section

Henry J. Buelt (M), Norman Eugene Danielson (J), William T. Huddle (J), Franklin J. Kern (A).

Salt Lake Group

Cliff H. Dunn (A).

Southern California Section

Henry L. Clark (M), Arthur P. Hayt (A), Herman Heidt (J), James Theodore Hird (M), Harrison W. Holzapfel (M), Frank Kurtis (M), Paul John Lansing (M), Herbert F. Sammons (M), Richard R. Shade (J), Charles Haldane Swan (J), Rhys O. Williams (J).

Southern New England Section

Hayden D. Alexander (A), Robert W. Dixon (J), William J. Finnell (J), J. R. Glynn (J), Peter J. Krones (J).

Spokane Intermountain Section

Lyman A. Powell (A).

Syracuse Section

Latham Thomas Winfree (A).

Texas Section

George H. Eaton (A), Walter H. Hutchcraft (A).

Twin City Section

Joseph J. Hite (M).

Virginia Section

Donald Haig Hinshelwood (A).

Washington Section

Loris N. Mouser (A), James Sterling Stelzer (J), Albert Fleming Thompson (A), Robert Ernest Samuel Thompson (A).

Western Michigan Section

D. A. Paull (M).

Williamsport Group

Harry J. Lavo (M).

Outside of Section Territory

William M. Cade (M), Dale S. Gronsdahl (J), Robert K. Heule (J), Johnnie Moran Lackey (A), James Grover Loudermilk (J), John Emerson Lovely (M), William Marcus Sanford (J), A. A. Schnorr (M), Franklin C. Walters (J), William Stanley Zartman (J), Steve Zelisko (A).

Foreign

Karl Gustav Ahlen (FM), Sweden; J. F. Buss (FM), Switzerland; Fikret Celtikci (FM), Turkey; John Arthur Cooper (FM), England; G. Depre (A), France; Richard George Grant (FM), Australia; Martin E. Holt (A), India; David Olof Johannes Lindgren (FM), Sweden; Mocherla Kanaka Raju (J), India; Nicholas Straussler (FM), England; Frank Alastair Wadsworth (FM), England.

Applications Received

The applications for membership received between March 10, 1949, and April 10, 1949 are listed below.

Baltimore Section

R. D. Austin, Jr., James Newton Berrett, Clarence M. Rusk.

British Columbia Group

John W. Booth, William J. S. Campbell, Ronald H. Carter.

Buffalo Section

Howard J. Jansen, Clement J. Turansky.

Canadian Section

Leonard A. Corn, Frank J. Crothers, Lee Tolmie Craig, Roy E. Taylor.

Chicago Section

William F. Fisher, Samuel T. Hobson, Joseph D. Kohutik, Leigh R. McCrea, Roger E. Pardon, Jerry Perlman, Eugene M. Sabo, William C. Shirley, Donn Sutton, Raymond S. Vicker, Arnold F. Zimmerman.

Cincinnati Section

C. Marvin Dorsey, Joseph Graber, Wayne D. Van Atta, Jr.

Cleveland Section

Newton D. Baker, III, George Fehlner.

Dayton Section

Kenneth C. Baker, Clarence H. Lorig, Herman J. Martin, Glen E. Mealy, Jr., John David Moeller, S. R. Prance, Joseph E. Scandling, Richard Kelsey Sullivan.

Detroit Section

Walter D. Baldwin, Hugh H. Benninger, Harold Raymond Boyer, Charles Norman Crossley, Philip R. Denham, Jr., John E. DeWald, Jack E. Gieck, Earle H. Gould, Gerald Grunow, James Henderson, Vernon E. Hense, Stanley R. Hood, Edward Paul Hornick, Alexander Hossack, F. E. Johnson, Cline Verne Larch, Charles W. Lesher, Robert S. MacDonald, Theodore J. Mategorin, Philip Mazziotti, J. Kelsey McClure, William L. Mitchell, Richard L. Pierce, E. R. Reeves, Robert W. Renwick, Floyd W. Robinson, Helene A. Rother, Hannes Adolf Ruesch, Clyde H. Schamel, Ronald W. Scheck, Frederic R. Smith, Kendall D. Taylor, Kenneth L. Vogt.

Indiana Section

Ram T. Tejwaney.

Kansas City Section

Francis Lee Spruill.

Metropolitan Section

L. M. Lawton, Archie R. McQuillen, Fred L. Woods, Jr.

Mid-Continent Section

Thomas H. Riggin.

New England Section

John Elijah Whipple.

Northern California Section

John L. Cooley, John C. Ellis, Paul C. Jaeger, E. Lloyd King, Frederic R. Watson, Roy W. Young, Jr.

Northwest Section

M. E. Earnheart, David L. Evans, Glenn A. Henry, Phil Howe, Richard P. Look, Fred William Purdom, W. H. Young.

Oregon Section

Howard I. Martine, Peter E. Piluso, Daniel O. Thornton.

Philadelphia Section

Robert John Liggett, John E. Murray, Donald P. Osterhout, Jr., Harry B. Peterson, Jr., Laurence J. Test, Donald L. Williams.

Pittsburgh Section

Edward J. Mulvihill, Raymond Talbot Sendell.

St. Louis Section

J. Kenneth Craver, Donald E. Essen, Kenneth R. Heilemann, Eugene Edward Wallace.

Southern California Section

Lee Stanley Akin, Robert E. Douglas, John Louis Harnack, Raymond E. McCallum, Robert Waudby Morris, Emanuel Patrick Plancey, Ford W. Spikerman, Robert J. Steiner, Billy L. Stocking, Walter A. Woron.

Southern New England Section

Harold E. Adams, John Jackson Blessley, John D. Frazee, Robert B. Powell, John Walter Wells.

Texas Section

Alvis Eugene Goodwin, J. T. Hurry, Richard Howard LeTourneau, Ralph E. Roberts.

Twin City Section

H. L. Harms.

Virginia Section

Robey W. Estes.

Wichita Section

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